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REVIEW OF ENGINEERING PROGRESS ABROAD

VOL. V. No. 12 DECEMBER, 1944

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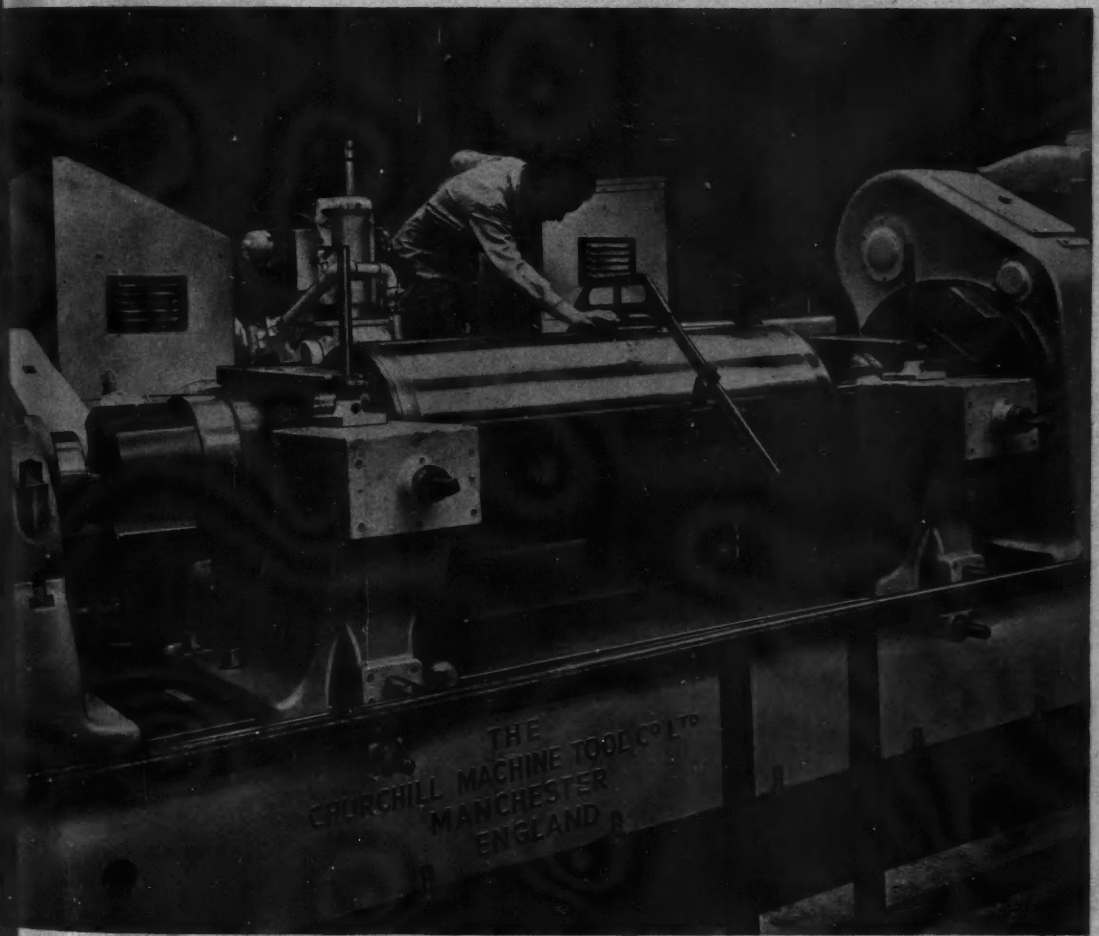

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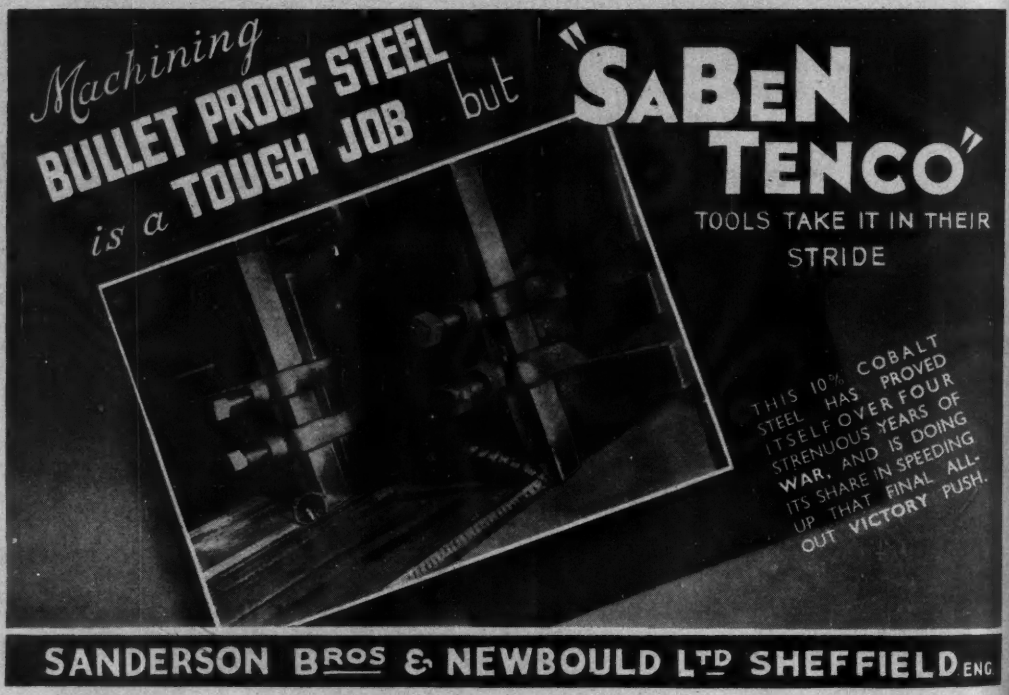
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CONTENTS

page

The Aerodynamic Turbine in Steel Plants 352
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 352.

Connecting Rod Acceleration 363
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 363.

Diagrams for Quick Determination of Angles Required for Laying out Sheet Metal Work .. 377
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 377.

Establishing Lists of Types in the Manufacture of Three-phase Motors 373
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 373.

Lathe Cutting Tests on Aluminium and Aluminium Alloys 355
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 355.

New Electric Drive for Rolling Mills 371
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 371.

Oerlikon Electro-Dynamometer Balance 369
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 369.

Plastic Bearings 366
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 366.

The Present Status of the Gas Turbine and its Economic Prospects 349
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 349.

Welding Conditions and Mechanical Properties of Spot-welded Aluminium Sheets 361
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 361.

Welding Machines with Condenser Discharge 377
" *The Engineers' Digest*," *Vol.* 5, *No.* 12, *p.* 377.

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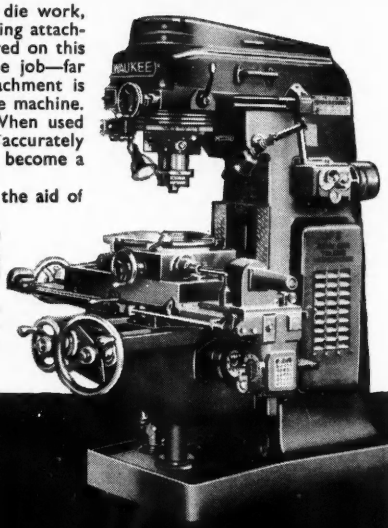

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LONDON BRISTOL BIRMINGHAM MANCHESTER LEEDS GLASGOW NEWCASTLE

THE PRESENT STATUS OF THE GAS TURBINE AND ITS ECONOMIC PROSPECTS.

By Dipl. Ing. H. PFENNINGER. (From *Schweizerische Bauzeitung*, Vol. 123, Nos. 24 and 26, 10th and 24th June, 1944, pp. 281-286 and pp. 307-311.

(*Continued from our previous issue*)

4. GAS TURBINES FOR BLAST FURNACES.

Gas turbines are especially well suited for this purpose (Fig. 11). Because of the small space they require they can be erected in the vicinity of blast furnaces. As water is not required, it is not involved in design considerations. The volume of air required for both turbine and blast furnace is delivered by a single compressor. The pressure ratio of the compressor is chosen for most favourable turbine design, while the air required for the blast furnace is obtained by bleeding the compressor at that stage which corresponds to the desired pressure for blast furnace operation. As the useful power is the energy of compressed air, the thermal efficiency can be stated best with reference to the adiabatic power. Thus, a direct comparison with steam plants becomes possible.

An advantage which should not be underestimated lies in the fact that the total volume of air compressed is great in relation to that required for the blast furnace. Therefore, the volume of blast air can be varied within wide limits.

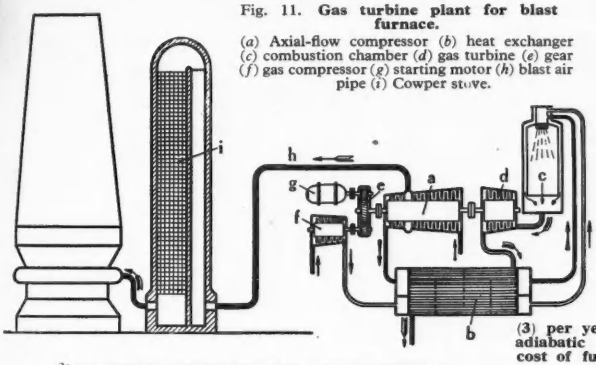
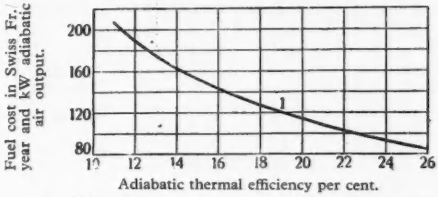


Fig. 11. **Gas turbine plant for blast furnace.**
(*a*) Axial-flow compressor (*b*) heat exchanger (*c*) combustion chamber (*d*) gas turbine (*e*) gear (*f*) gas compressor (*g*) starting motor (*h*) blast air pipe (*i*) Cowper stove.

The economic considerations involved in the design of such a plant are illustrated by Figs. 12 to 14. Fig. 14 shows that the greatest annual saving accruing from installation of a heat exchanger is obtained at an adiabatic thermal efficiency of 22 per cent. To compare this value with the effective efficiency of a steam plant at the clutch of the steam turbine it is necessary to divide it by the thermodynamic efficiency of the blast blower. Assuming 78 per cent for this efficiency, it gives a thermal efficiency of 28 per cent for an equivalent steam plant. Further comparison as to



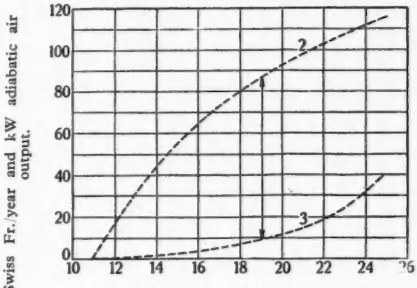
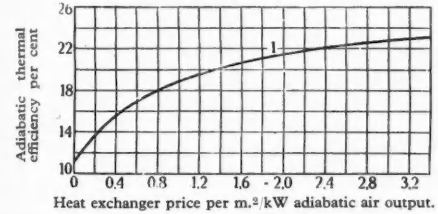


Fig. 13. **Fuel cost per year and kW adiabatic (1), saving in fuel cost due to employment of heat exchanger (2) and heat exchanger cost (3) per year and kW. adiabatic output as a function of adiabatic thermal efficiency. Fuel: blast furnace gas, cost of fuel 4.4 Rpf./10,000 kcal. Operation: [6000 hours per year.**



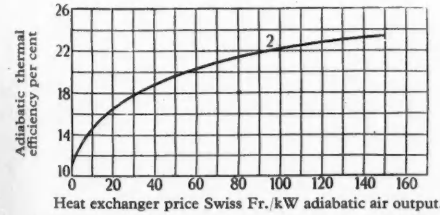


Fig. 12. **Thermal efficiency of a single-stage gas turbine for blast furnace as a function of heat exchanger surface (1) and as a function of heat exchanger price per kW adiabatic air output (2). The heat exchanger price has been taken as 40 Sw. Fr. per sq. m.**

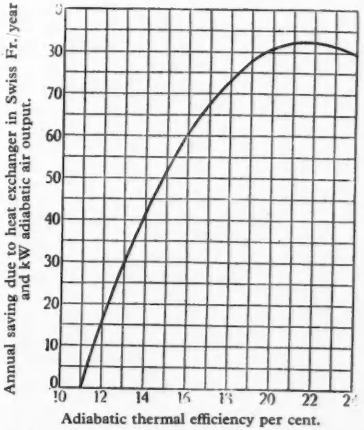


Fig. 14. **Annual saving due to installation of heat exchanger as a function of thermal efficiency. (Difference between curves 2 and 3 in Fig. 13.)**

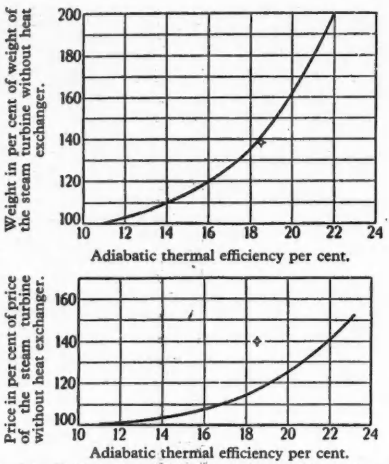


Fig. 15. **Cost and weight comparison with a steam turbine for blast furnace air supply. Blast air of 100,000 kg./hr. Air pressure 2.2 atmos. abs.**
—— Gas turbine. ✧ Steam turbine.

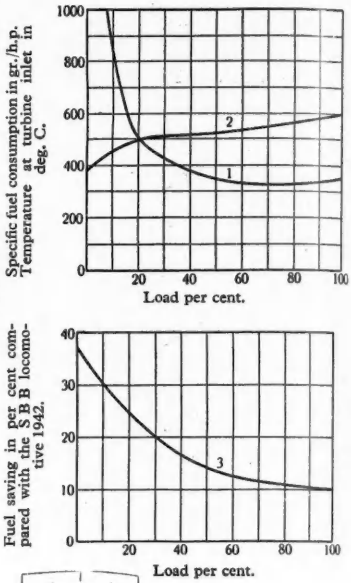


Fig. 16. **Specific fuel consumption (1) and temperature (2) at turbine inlet as a function of load of a gas turbine for 2500 h.p. Fuel saving (3) of the new gas turbine of 2500 h.p. as a percentage of the consumption of the first S B B locomotive.**

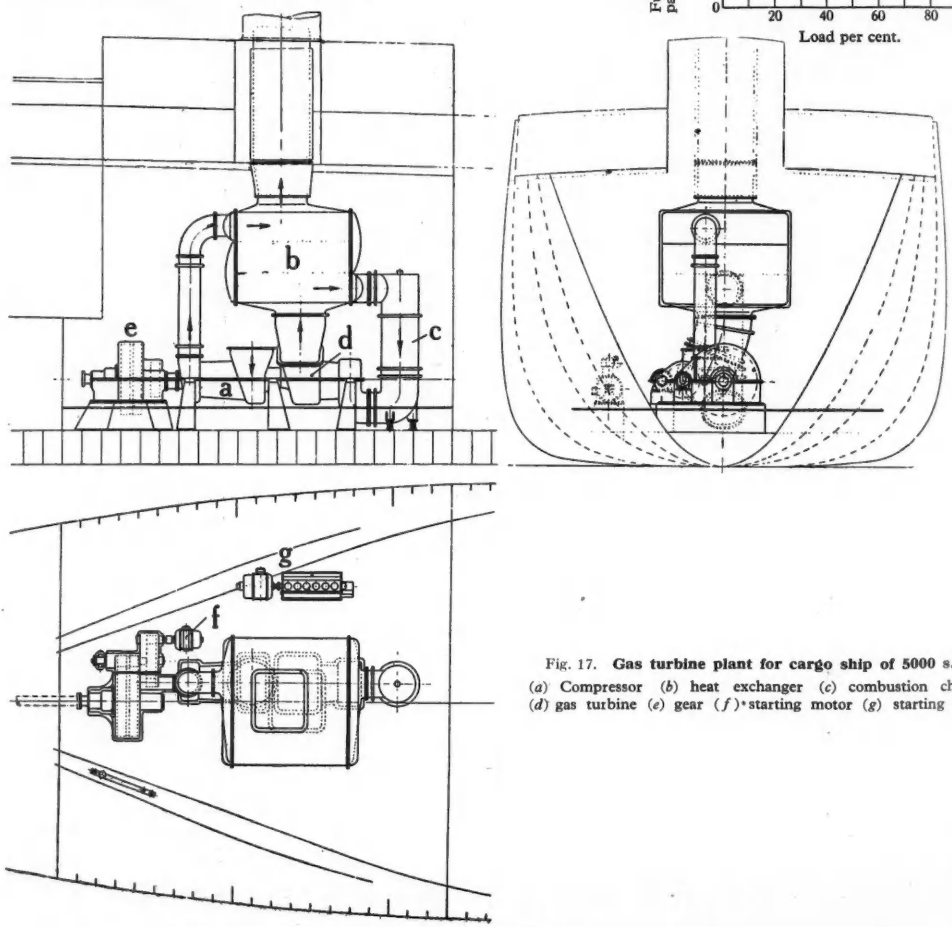


Fig. 17. **Gas turbine plant for cargo ship of 5000 s.h.p.**
(*a*) Compressor (*b*) heat exchanger (*c*) combustion chamber (*d*) gas turbine (*e*) gear (*f*) starting motor (*g*) starting diesel.

weight and price of steam plant and gas turbine plant is shown in Fig. 15.

5. THE GAS TURBINE FOR LAND OR MARINE TRANSPORTATION.

The Swiss Federal Railways (SBB) have had a gas turbine locomotive in use since May, 1943, which has proved satisfactory. A further development of this unit is under construction, but of 2500 h.p., occupying the same space and being of nearly the same weight. The specific fuel consumption of this unit is shown in Fig. 16.

The gas turbine with heat exchanger is also well suited for marine cargo application. The thermal efficiency should reach 24 to 27 per cent. For this, a larger heat exchanger is required which, however, works out economically as the fuel saving would decrease the weight of fuel carried and the radius of action would be increased. Table II of the original article gives a comparison with steam turbine propulsion units. The simplicity of the gas turbine cargo ship is most clearly shown by the absence of the boiler with its auxiliaries and of the costly preparation of water. Fig. 17 shows a cargo ship with gas-turbine drive of 5000 s.h.p. As can be seen, the power plant can be built very compactly, and placed well towards the stern of the ship, at the least valuable space for cargo. In addition, the propeller shaft is shortened, resulting in further simplification and space saving.

Although the thermal efficiency of the Diesel engine is 38 per cent, the gas turbine is superior in this field as well. This superiority is due to the high price of diesel oil (see Table I) which, for example, would result in equal fuel cost for the two types of plants if the thermal efficiency of the gas turbine were as low as 19 per cent. In addition, the cost of lubrication is higher with diesel engines. According to Chapman,* the cost of lubricating oil for diesel locomotives is 20 to 30 per cent of the fuel cost. An economy diagram is shown in Fig. 18, the dotted lines with arrowheads explaining its use.

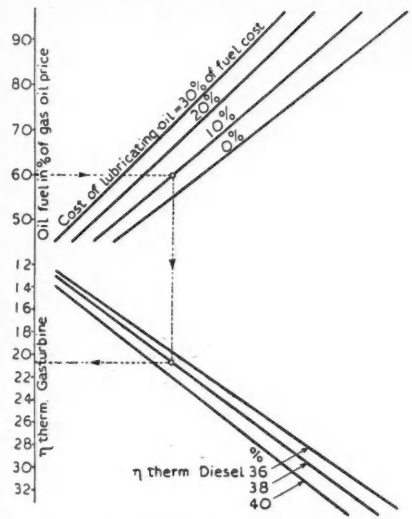


Fig. 18. **Nomogram for comparison of the economy of a diesel engine with that of a gas turbine.**

6. THE LIGHT-WEIGHT GAS TURBINE FOR TRANSPORT PURPOSES.

If the tests which are now proceeding prove satisfactory, it should be possible in the near future to build gas turbine plants with a unit weight of 3 to 4 kg./h.p. The space requirements of such a plant are about 30 cu. m./1000 h.p. Fig. 19 shows a light-weight plant for 5000 h.p. which is suitable for boosting battleships. In this case the fuel consumption is of minor importance; what matters, is small space and light weight.

For aircraft installation this type of plant is not yet suitable. What appears to have developed to the experimental stage now is a combination of piston engine and gas turbine as shown in Fig. 20. The piston engine (2) drives the compressor (1), the compressed air being passed through the combustion chambers (3) into the turbines (4) which drive the propellers. The piston engine (2) is supercharged by compressor (1). By mixing the exhaust from engine (2) with the hot gases and expanding it through the turbines to atmospheric pressure, additional kinetic energy is gained for the propulsion of the aircraft. The unit weight of such a plant is 1 kg./h.p.; its thermal efficiency is around 28 per cent.

7. CONCLUSION.

Gas turbines have proved themselves reliable in operation over a number of years. The first units were destined for the Houdry refining process where large volumes of compressed air are required. These plants have worked since 1938 without interruption. Twenty-one units have been delivered up to 1942, of which nineteen are in operation in the United States.

In contradistinction to piston engines, the gas turbine

* E. E. Chapman, "Steam versus Diesel-Electric Power," *Railway Age*, Vol. I, 1941, No. 4, July, 1926.

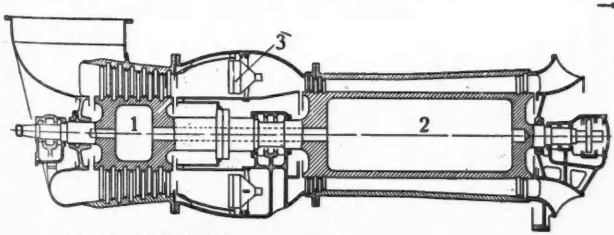


Fig. 19. **Light-weight gas turbine plant.**
(1) Gas turbine. (2) compressor. (3) combustion chamber.

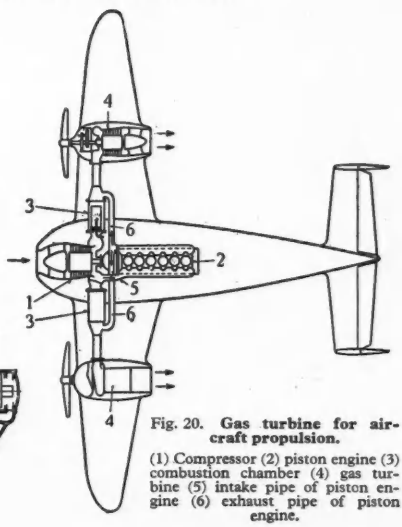


Fig. 20. **Gas turbine for aircraft propulsion.**
(1) Compressor (2) piston engine (3) combustion chamber (4) gas turbine (5) intake pipe of piston engine (6) exhaust pipe of piston engine.

has no elements subjected to wear such as ventiles, piston rings and cylinder bushes. There are no reciprocating masses either, therefore vibration is absent. The figure for excess air with a combustion engine is about 1.5, while for a gas turbine at a thermal efficiency of 28 per cent it is 16. This high excess of air excludes the danger of corrosion of the turbine blades. Beyond the field of present-day application, the gas turbine holds the promise of great potentialities. The coal-fired gas turbine, for instance, is now in the experimental stage. Apart from the possibility of burning coal dust in the combustion chamber, the employment of a gas generator appears to be the right solution. The gas generated is then burned in the combustion chamber, so avoiding slag and ash troubles.

THE AERODYNAMIC TURBINE IN STEEL PLANTS.

By C. KELLER and R. RUEGG. (From *Stahl & Eisen*, Vol. 64, No. 13, March 30th, 1944, pp. 201-206).

AERODYNAMIC turbines are a promising means for improving and simplifying steel works' practice, as they conform particularly well with the prevailing exigencies of reliability, simplicity, small space requirements, and low heat consumption. As the aerodynamic turbine operates with clean air as working medium instead of flue gas, an air preheater is required, the design of which is similar to that of hot blast heaters. A combination of hot-blast heater and turbine air-heater therefore offers considerable advantages both with regard to layout of plant, and thermal economy.

The replacement alone of steam drives by aerodynamic turbines, results in considerable simplification of plant, eliminating the need for boiler water treatment plant and a considerable number of auxiliaries. It has been objected that the achievement of high gas-turbine efficiency is generally conditional upon the employment of large and bulky heat exchangers for the utilisation of waste heat contained in the turbine exhaust. This argument, however, is largely inapplicable to the aerodynamic turbine. Owing to the use of a clean working medium, i.e., air, at elevated pressure, the heat exchanger surfaces of the air turbine are only a fraction of those required for the open gas-turbine in which atmospheric pressure must remain the lower pressure level (see Fig. 1).

For a 300 kW blast turbo-blower plant, the dimensions of the components will be approximately as follows :

Turbines : High pressure part, 4 stages, mean diameter 400 mm. ; Low pressure part, 5 stages, mean diameter 700 mm.

Circuit air compressor of the axial flow type : mean diameter 380 mm. at admission, and 300 mm. at delivery end.

Heat exchanger : Surface area 1200 sq. m. ; diameter 1.2 m. ; length 10 m.

The dimensions of the air heater proper of present-day design approximate to those of a steam boiler of equivalent capacity. But it must be kept in mind that the air heater and hot-blast heater may be combined in a single unit. The following tentative lay-outs show the various possibilities of employing aerodynamic turbines in steel works ; but these lay-outs by no means comprise a complete list of applications.

In the layout shown in Fig. 2, the blast blower is driven by an aerodynamic turbine, the air heater of which is combined with the blast heater. For the sake of greater clarity, approximate pressures and temperatures are indicated in this diagram. The special arrangement of the flow circuits has been chosen in order to secure the highest possible fuel economy. Thus, for instance, the flue gas path in the air preheater is arranged in two passes. The first, or high temperature pass, serves for heating the working air of the turbine cycle to its final temperature, while the second, low temperature pass, is used for heating the blast delivered by the blower. In Fig. 2 the working air heater is shown in the form of a straight tube bundle arranged in the first flue gas pass, but its heating surface may also be of the radiant type lining the walls of the furnace (Fig. 5).

Heretofore, the low pressures of 1-2 kg. per sq. cm. prevailing in tubular blast heaters made it necessary to employ comparatively high air velocities, and accordingly high pressure drop in order to achieve the rates of heat transfer required and to prevent overheating of the tubes. In this respect the higher pressure levels of the working air circuit—varying from 20-30 kg. per sq. cm.—is of especial advantage, as the heat transfer rate increases with the factor $(cp)^{0.75}$ where c is the velocity and p the pressure. In other words, the heat transfer coefficient increases with the pressure to the same extent as with the velocity of flow. It is, therefore, an easy matter to obtain sufficient cooling of the air heater

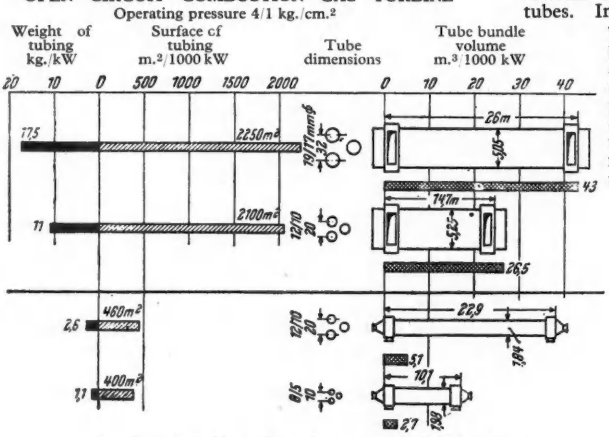


Fig. 1. **Comparison of heat exchanger sizes.** Sizes of heat exchangers for equivalent 12,000 kW open circuit gas turbine and aerodynamic turbine. Compression ratio $p_1/p_2 = 4$. Because of the higher pressure of the working medium, the size and weight of the tube bundle of the aerodynamic turbine are one-sixteenth that of the open gas circuit turbine.

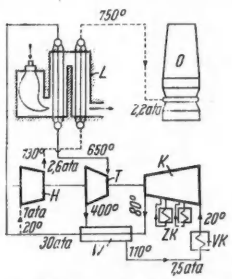


Fig. 2. **Blast supply plant equipped with aerodynamic turbine drive and combined blast and circuit-air heater.**

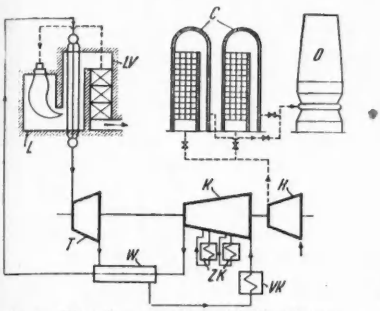


Fig. 3. **Blast furnace plant with blast heater and turbo blower driven by aerodynamic turbine plant.**

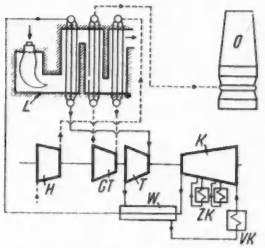


Fig. 4. **Blast compressor driven by back-pressure aerodynamic turbine combined with aerodynamic power plant.**

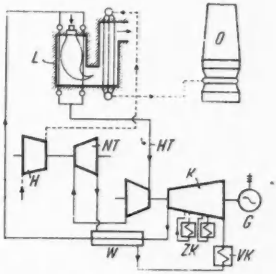


Fig. 5. **Blast compressor with aerodynamic drive with separate shafts and equipped with tubular air heater.**

O Blast furnace ; *C* Cowper stove ; *H* Blast compressor ; *L* Blast furnace gas fired air heater for blast and circuit-air ; *LV* Combustion air preheater ; *GT* Back-pressure turbine ; *T* Aerodynamic turbine drive ; *HT* Aerodynamic high pressure turbine ; *NT* Aerodynamic low pressure turbine ; *K* Circuit-air compressor ; *ZK* Intercooler ; *VK* Pre-cooler ; *W* Heat exchanger ; *G* Generator for excess power.

tubing arranged in the hottest gases. On the other hand, the tubes of the blast heater are placed in the second gas pass where lower gas temperatures prevail, thus facilitating limitation of the tube temperatures to the maximum permissible level.

A further advantage of this air heater arrangement is that the heat from the flue gas is absorbed by air of relatively low temperature, thus permitting a considerable reduction in the flue gas temperature with a correspondingly small uptake loss. Combustion air preheaters, as they are used in power plant practice for the extraction of waste heat from the stack gases, can therefore either be omitted altogether, or be considerably reduced in heating surface.

The tubular blast heaters and an experimental circuit air heater built so far has proven the feasibility of high temperature air heating. Where an existing plant is equipped with a checker-work type regenerative heater (Cowper stove), a first step in its modernisation may consist in equipping the blast furnace blower with an aerodynamic turbine drive. A scheme of this kind is illustrated in Fig. 3. At a later date, a tubular blast heater may be added to replace the Cowper stove. The plant shown in Fig. 2 represents the simplest arrangement of its kind. But wherever industrial process heat is required, the question arises whether a better thermal efficiency cannot be achieved by resorting to by-product power generation, the exhaust of the back-pressure operated prime mover serving to meet the process heat requirements.

As a final hot blast temperature of 750 deg. C. is to be attained, it is not possible to meet the entire heat requirements by waste heat extraction, but a certain portion can at least be obtained in the manner outlined in Fig. 4. Here the compressor driven by the aerodynamic turbine produces a higher pressure than needed for the hot blast. This compressed air is heated in the first stage of the blast heater and is subsequently expanded in the aerodynamic turbine *GT* coupled with the compressor. As this turbine is too small to meet the entire power requirements of the compressor, additional power is obtained from the aerodynamic power turbine *H* which is also coupled to the compressor and turbine *GT*. The air discharged from turbine *GT* is reheated in the second stage of the blast heater to the temperature required at the blast furnace. This arrangement has the advantage that the specific heat requirements per unit volume of hot blast are less by six per cent than those of an equivalent plant laid out according to Fig. 2.

The heat requirements and operating data of plants Fig. 2 and Fig. 4 are listed in Table I., the thermal efficiency of the aerodynamic turbine being assumed as 30 per cent. Higher thermal turbine efficiencies will be reflected by corresponding decreases in the heat consumption rates. The figures given show that plants of this type are competitive with the most efficient gas engine drives available. Specific heat consumption rates per unit volume of hot blast are largely dependent upon the back pressure chosen for the aerodynamic turbine. It will be seen from Fig. 6 that for a hot blast condition of 750 deg. C. at 2.2 kg. per sq. cm., lowest heat consumption will be obtained by operating the aerodynamic turbine with a back pressure of about 6 kg. per sq. cm. The choice of a somewhat smaller pressure appears, however, indicated, as this decreases the compression ratio of compressor and back-pressure turbine which in turn leads to lower the cost of the plant.

TABLE I.

HOT BLAST PLANT WITH AERODYNAMIC TURBINE DRIVE.

	Plant according to Fig. 2	Plant according to Fig. 4
(1) *Blast Blower H*		
Blast quantity, kg/hr.	100,000	100,000
Final temp. of blast, deg. C.	750	750
Blast pressure at blast furnace, kg./sq. cm.	2.2	2.2
Compression ratio	2.6	4.3
Adiabatic work of compression, kcal/kg.	22.1	36.5
Compression efficiency, per cent	85	85
Air temp. at blower outlet, deg. C.	130	202
Power consumption of blower, kW	3,020	4,995
(2) *Back-pressure Aerodynamic Turbine GT*		
Quantity of air, kg./hr.	—	100,000
Pressure at turbine inlet, kg./sq.cm.	—	4.0
Temperature at turbine inlet, deg. C.	—	650
Pressure at turbine outlet, kg./sq.cm.	—	2.4
Temp. at turbine outlet, deg. C.	—	537
Pressure ratio	—	1.67
Work of expansion, kcal/kg.	—	30.2
Turbine efficiency, per cent	—	88
Output of turbine, kW	—	3,085
(3) *Heat consumption of Drive and Blast Heater*		
Additional power supply, kW nett	3,020	1,910
Corresponding heat consumption, 10^6 kcal/hr.	8.66	5.48
Heat consumption for heating of blast, 10^6 kcal/hr.	18.75	20.25
Total	27.41	25.73 10^6 kcal/hr.
Heat consumption per NPT cub.m. of blast, kcal	354	332
(4) *Aerodynamic Power Unit*		
Temperature at turbine inlet, deg. C.	650	650
Upper circuit pressure p_1 at max. load, kg./sq.cm.	30.0	30.0
Lower circuit pressure at max. load, kg./sq.cm.	7.5	7.5
Air circulation, kg./hr.	109,000	69,000
Gross power output of turbine T, kW	7,550	4,770
Power consumption of compressor K, kW	4,530	2,860

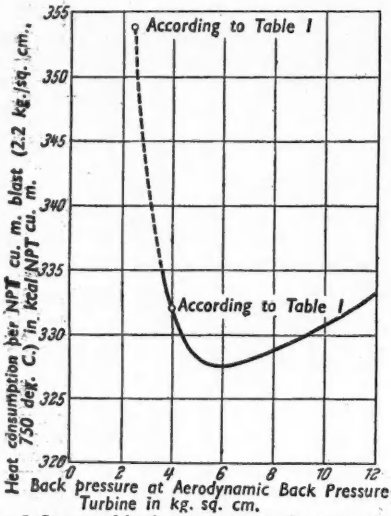

According to Table I
According to Table I
Heat consumption per NPT cu. m. blast (2.2 kg./sq. cm., 750 deg. C.) in kcal/NPT cu. m.
Back pressure at Aerodynamic Turbine in kg. sq. cm.
Back Pressure

Fig. 6. Influence of back-pressure upon heat consumption of hot blast plant with blast blower driven by aerodynamic back-pressure turbine and combined with aerodynamic power plant.

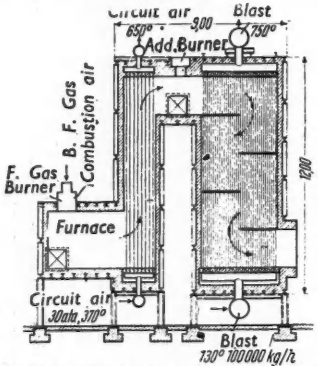


Fig. 7. Combined circuit-air and blast heater for 100,000 kg./hr. blast.

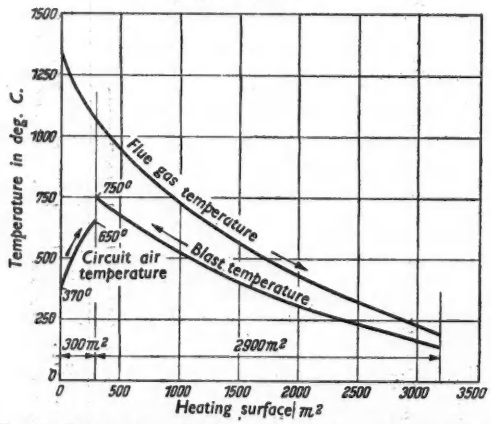


Fig. 8. Temperature distribution in surface of combined circuit-air blast heater for 100,000 kg./hr. blast and 109,000 kg./hr. circuit-air.

The subjoined data, therefore, are based on a back-pressure of 4 kg. per sq. cm.

Actually, it will be preferable to equip plants Figs. 2, 3 and 4 with high and low pressure turbines as shown in Fig. 5. In this arrangement, the low pressure turbine drives the blast compressor at variable speed corresponding to blast requirements, while the high pressure turbine serves to drive the high speed compressor of the aerodynamic power unit. Output of the latter is controlled by adjusting the compression ratio of the circuit.

The sectional view of a combined circuit-air and blast heater for 100,000 kg./hr. is given in Fig. 7, while the data appertaining are given in Table II.

TABLE II.

COMBINED AIR AND BLAST HEATER ACCORDING TO FIG. 7.

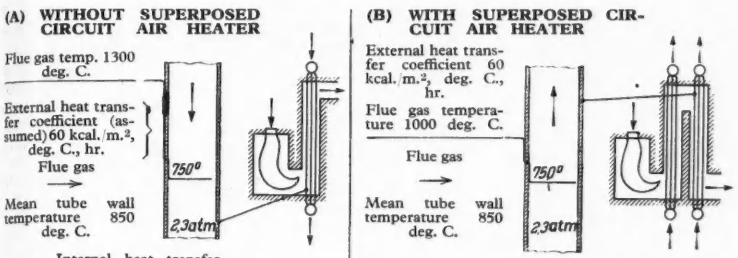
(1)	Calcrific value of blast furnace gas, kcal/NPT cu.m.	1,000
(2)	Efficiency of air and blast heater, per cent	85
(3)	Air quantities : Blast, kg./hr.	100,000
	Working air, kg./hr.	109,000
(4)	Heat quantities (according to Table I)	
	For power generation, 10^6 kcal/hr.	8.66
	For blast heating, 10^6 kcal/hr...	18.75
	Total	27.41 10^6 kcal/hr.
(5)	Heat quantities passing through heating surfaces :	
	Circuit air, 10^6 kcal/nr.	7.35
	Blast, 10^6 kcal/hr.	15.95
	Total	23.3 10^6 kcal/hr.
(6)	Blast furnace gas, NPT cu.m./hr.	27,410
(7)	Combustion air (20 per cent excess air), NPT cu.m./hr.	27,600
(8)	Flue gas (20 per cent excess air), NPT cu.m./hr. ..	52,000
(9)	Uptake temperature, deg. C., approx.	185
(10)	Heating surface :	
	(a) for circuit air, sq. m.	300
	(b) for blast heating, sq.m. ..	2,900
	Total	3,200 sq.m.

The furnace proper is of small dimensions, as the use of gaseous fuel permits the employment of high rates of heat release. The heating surface of 300 sq. m. for heating the circuit-air is arranged in the first pass of the heater, its surface area amounting to only about one-tenth of that required for heating the blast. Between first and second pass of the heater an additional burner is provided which enables independent adjustment of the hot blast temperature. The temperature distribution throughout the heating surfaces of the combined circuit-air hot blast heater is charted in Fig. 8. It is also seen that the uptake temperature is low enough to obviate the need for a special combustion air preheater.

It has already been mentioned that considerable advantages with regard to heat transfer accrue from the employment of compressed air. This improvement in the coefficient of heat transfer between the air and the tube wall is so pronounced, that with small air velocities and correspondingly low pressure losses, tube wall temperatures remain within the range of commercially available alloy-tube materials.

Analogous considerations affect the choice of the air velocity in the hot blast heating tubes, as an excessive pressure drop would call for an unduly large power consumption of the blower, and an excessive heat consumption per unit volume of hot blast. The beneficial effects in this respect which accrue from placing the circuit-air heater in front of the blast heater, are evidenced by the comparative data given in Fig. 9. Here it is seen that the addition of the circuit-air heater reduces the pressure drop requirements in the most exposed part of the blast heater from 1,090 mm. w.g. to 65 mm. w.g.

This comparison is based on the assumption of a heat transfer coefficient of 60 kcal per sq. m. per deg. C. per hour for both arrangements. In the case of the blast heater alone it would, of course, be possible to lower this coefficient by using a lower gas velocity; but it would be difficult to achieve a coefficient of less than


(A) WITHOUT SUPERPOSED CIRCUIT AIR HEATER
Flue gas temp. 1300 deg. C.
External heat transfer coefficient (assumed) 60 kcal./m.², deg. C., hr.
Flue gas
Mean tube wall temperature 850 deg. C.
750°
2.3atm
Internal heat transfer coefficient αi = 309 kcal./m.², deg. C. hr.
Air velocity W = 178 m./sec.
Pressure drop Δp = 1090 mm. WS/m.
(B) WITH SUPERPOSED CIRCUIT AIR HEATER
External heat transfer coefficient 60 kcal./m.², deg. C., hr.
Flue gas temperature 1000 deg. C.
Flue gas
Mean tube wall temperature 850 deg. C.
750°
2.3atm
αi = 103 kcal./m²., deg. C., hr.
W = 41 m./sec.
Δp = 65 mm. WS/m.

Therefore, Pressure Drop with A = 17 times Pressure Drop with B.

Fig. 9. **Comparative pressure drops per metre length of tube at outlet end of blast heater tubes.**

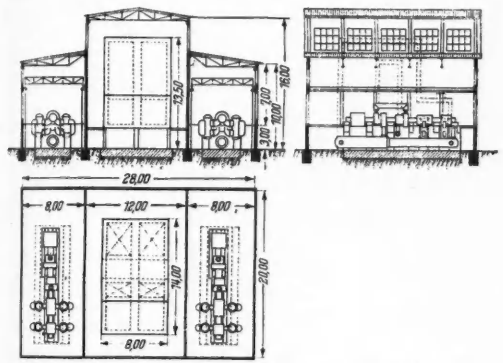


Fig. 10. **Tubular blast heater and blowing plant with aerodynamic turbine drive.**
Blast : 2 × 100,000 kg./hr., temperature 750 deg. C.
Blower capacity : 2 × 3,000 kW.

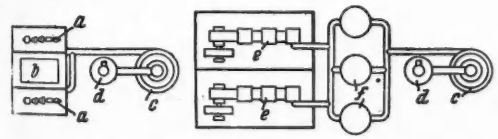


Fig. 11. **Space requirements of blower plant with tubular blast heater and aerodynamic turbine drive *versus* plant with Cowper stoves and reciprocating blowing engines for 100,000 kg./hr.**
(a) Blower with aerodynamic turbine drive
(b) Tubular blast heater
(c) Blast furnace
(d) Dust separator
(e) Gas engine blower
(f) Cowper stove.

30 kcal. per sq. m. per deg. C. per hour, since the required increase in tube spacing would increase the heat transfer by radiation. But even so, the blast heater alone, in its most exposed parts, would have a specific pressure drop three times that of the combined apparatus.

The combined blast heater, the blast blower, and the aerodynamic drive will best be arranged in one single building as shown in Fig. 10. This lay-out provides for two independent sets, each being capable of supplying 100,000 kg. per hour of hot blast at 2.2 kg. per sq. cm. pressure and 750 deg. C. temperature. With a compact lay-out of this type, piping connections are short and plant operation is easy. The saving in space achieved with the use of the aerodynamic unit as compared to the usual type of plant equipped with reciprocating blowers and Cowper stoves, is evidenced by Fig. 11. The layout of an entire blast furnace plant equipped with aerodynamic turbine is given in Fig. 12. Each of the two blast furnaces is assumed to require 50,000 kg. of hot blast per hour. In order to provide a 100 per cent reserve capacity, it is proposed to instal two independent hot blast supply plants, each with a capacity of 100,000 kg. per hour. Modern blast furnace plants are frequently equipped with blast furnaces having daily outputs as large as 1,000-1,200 tons, and these require blowers of correspondingly large capacity. Thus, for instance, two turbo blast compressors each of 300,000 kg. per hour capacity, with a delivery pressure of 1.8 kg. per sq. cm. and equipped with a 10,000 kW drive, were supplied by the Escher Wyss company to a blast furnace plant in India.

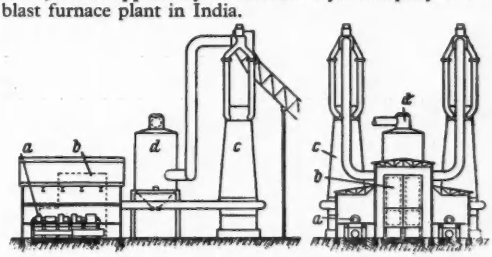


Fig. 12. **Blast furnace with compressor plant driven by aerodynamic turbine. Two blast furnaces of 50,000 kg./hr. blast requirements. Two blowers of 100,000 kg./hr. each (Full reserve capacity).**
(a) Blower with aerodynamic turbine (b) Tubular blast heater
(c) Blast furnace requiring 100,000 kg./hr. blast
(d) Dust separator (e) Gas engine blower.

LATHE CUTTING TESTS ON ALUMINIUM AND ALUMINIUM ALLOYS.

By E. von Burg. (From *Schweizer Archiv*, Vol. 10, No. 6, June, 1944, pp. 161-177.)

To the knowledge of the author, technical literature contains no useful references to systematic investigations of the cutting forces obtaining in the cutting of aluminium with high rates of feed and high cutting speeds. Knowledge of the cutting forces is, however, important because they serve as a significant criterion of the machinability of a material, the other criteria being the surface condition of the machined material and the life of the cutting tool. The cutting forces are also important in connection with the power requirements of the lathe.

The main purpose of the present investigation was the determination of the component cutting forces in relation to the cutting speed. The cutting forces were computed from the electric energy consumption of the lathe. Tables I and II show the investigated alloys in order of their cutting forces by taking a depth of cut of 2 mm. and a feed of 0.3 mm. as a basis.

Note should be taken of the favourable influence exerted by high cutting speed upon chip formation—that is, upon the quality of the machined surface. For the investigation of the machinability of a given material four basic test procedures (partly complementary to each other) are available, viz. :—

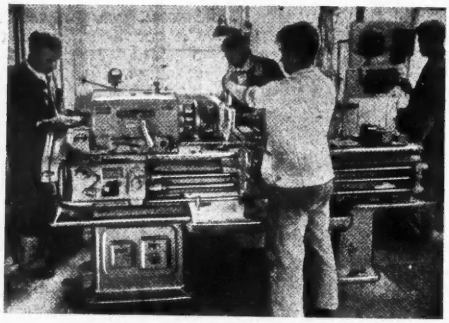


Fig. 1. **Lathe used for test.**

(1) Determination of the cutting forces reacting upon the single-point tool.
(2) Determination of tool wear.
(3) Determination of surface quality of the machined material.
(4) Determination of cutting temperatures.

The tests were made for practical rather than for theoretical purposes, and are intended to establish optimum machining conditions. The lathe employed for the investigation, which is shown in Fig. 1, was originally fitted with an electric motor of 3 h.p., but had to be replaced by a 6.5 h.p. motor. It was first intended to measure the cutting forces direct by means of electrical dynamometers fitted with dial indicators, but these were unsuitable because of vibrations prevailing with large cutting depths. It was therefore preferred to compute the cutting forces from the electrical power consumption of the lathe.

TABLE I.—MATERIAL AS SUPPLIED, OR IN SOFT CONDITION

	Average main cutting force, kg.			
	$v = 100$	300	500	700 m./min.
Casting : Pure Al ; Al + 5 per cent Cu Al + 10 per cent Cu Anticorodal (Al-Si-Mg)	34–41	29–32	28–31	27–30
Casting : Avional DTi (Al-Cu-Mg) Casting and Pressing : Avional S (Al-Cu-Mg) Pressing : Anticorodal (Al-Si-Mg)	39–44	32–39	29–38	—

TABLE II.—MATERIAL IN HARDENED CONDITION

Alloy	Average main cutting force, kg.			
	$v = 100$	300	500	700 m./min.
Casting : Al + 5 per cent Cu Al + 10 per cent Cu Casting and Pressing : Anticorodal (Al-Si-Mg)	34–37	31–32	30–31	29–30
	38–44	34–37	32–35	30–33
Casting : Avional DTi (Al-Cu-Mg) Casting and Pressing : Avional S (Al-Cu-Mg)	44–53	38–46	36–45	—

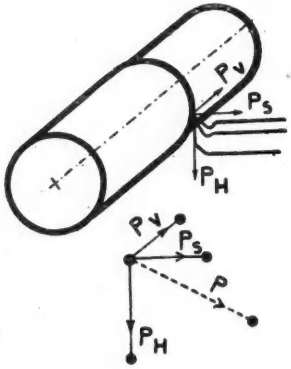


Fig. 2. **Cutting force components.**

Referring to Fig. 2, the cutting force P exerted upon the tool is the resultant of the three components Pv = longitudinal force, Ps = radial force, and P_H = tangential force. The velocity component associated with the longitudinal force is the feed rate of the tool, which from the aspect of this investigation is negligible. It amounts to 0.0074 m./sec. at 1,486 r.p.m., the maximum speed employed. The corresponding power consumption therefore may be assumed as nil. As no velocity component exists in the case of the radial force, the corresponding power consumption is also nil. Practically, the power absorption is therefore solely determined as the product of the tangential force P_H and its coordinated speed component v which is the peripheral speed of the work piece. The power absorption is thus given by $L = P_H v$ (kg. m./sec.).

With an efficiency η, the power absorption of the lathe is given by $L = N \cdot \eta$, where N is the electrical energy consumption of the motor. Since $N \cdot \eta = P_H \cdot v$, it can be written $P_H = N \cdot \eta / v$ (kg.). If n is the number of revolutions of the spindle per minute, and D is the diameter of the work piece, the cutting speed will be given by $v = D\pi n/60$ (m./sec.). The efficiency of the lathe was determined at spindle speeds of n = 186, 371, 528, 742, 1,058 and 1,486 r.p.m. The relationship between lathe efficiency and power absorption at different spindle speeds is charted in Fig. 3.

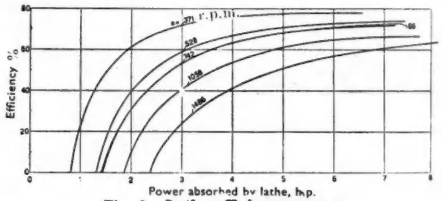


Fig. 3. **Lathe efficiency curves.**

In determining the most suitable shape of the cutting tool from the aspects of highest cutting efficiency and surface quality, the tool angles (Fig. 4) must be chosen from the viewpoints of *large depth of cut, small rate of feed, and high cutting speed.* The tool angles recommended for high speed steel are listed below.

Brinell hardness of light alloy to be machined	Cutting edge angle, degs.		
	α	β	γ
≤ 50	6–10	30–35	54–45
50–80	6–10	35–45	49–35
> 80	6–10	45–50	39–30
High Si alloy	6–9	62–66	22–15
Alloy for automatics	6–10	79–80	5–0

For sintered carbide tipped tools the following angles may be used : α = 6–10, β = 65–80, γ = 19–0.

Diamond tools have proven highly suitable for the dressing of aluminium work pieces. They are specially suited to mass production purposes and are outstandingly useful in the machining of Si-alloys.

The investigations were conducted with cobalt steel, the tool angles being α = 8 deg., β = 45 deg., and γ = 37 deg. with an entering angle of 90 deg. Since it was found that the application of a cutting fluid

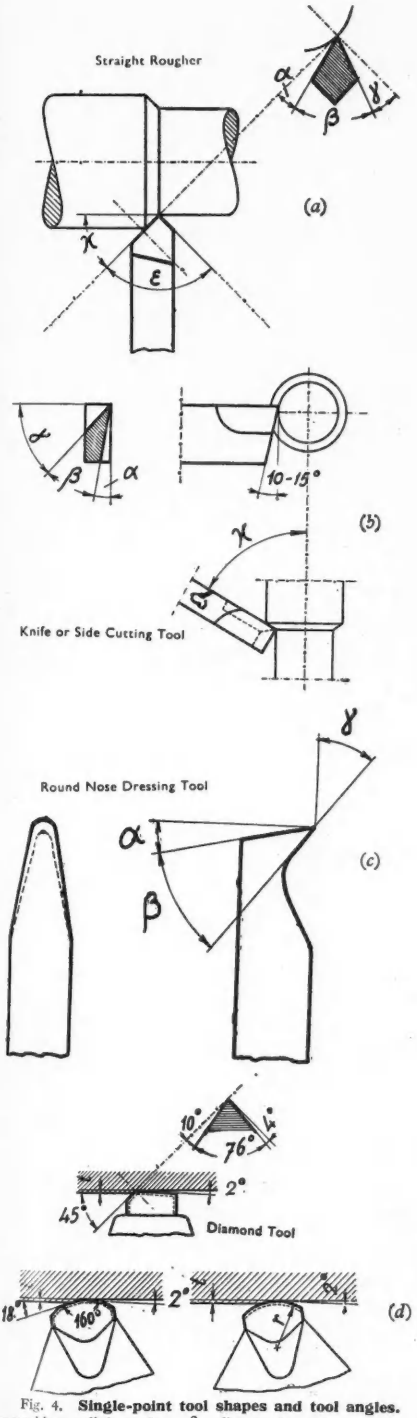


Fig. 4. **Single-point tool shapes and tool angles.**
α = working relief angle; β = lip angle; γ = rake angle; ϵ = nose angle; κ = entering angle.

remained without noticeable influence upon the cutting force, all tests were conducted without cutting fluid.

The work pieces were thick-walled tubes of 200 mm. external and 90 mm. internal diameter, the respective compositions of the pieces being as follows :—

	Si	Fe	Mn	Mg	Cu	Ti
Pure Al	0.1	0.4	—	—	—	—
Al + 5 per cent Cu	0.2	0.5	—	—	4.7	—
Al + 10 per cent Cu	0.2	0.6	—	—	10.4	—
Al–Si–Mg:						
Anticorodal ..	0.9	0.2	0.3	0.6	0.1	traces
Al–Cu–Mg:						
Avional DTi ..	0.3	0.3	0.5	0.5	3.9	0.1
Avional S.. ..	0.7	0.3	1.1	0.9	5.1	—

Prior to machining, samples were cut off for examination of the macrostructure.

The tangential or main cutting force was determined for different speeds with depths of cut of 1, 2 and 3 mm., the feed rate being 0.3 mm. The data obtained were entered in "Cutting speed" *v.* "Cutting force" charts, in which the scatter of the data is indicated by maximum and minimum range curves. Mean values are shown by curves drawn in broken line.

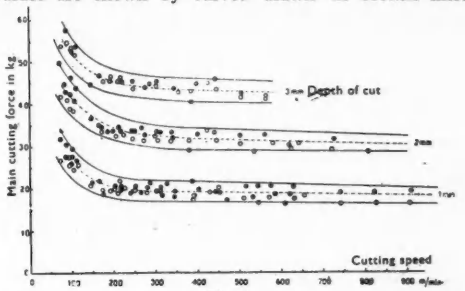


Fig. 5.
Machining test of Al + 5 per cent Cu casting "as supplied."

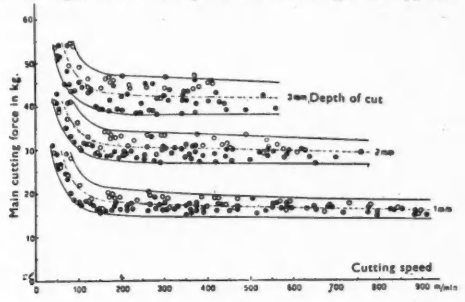


Fig. 6.
Machining test of Al + 10 per cent Cu casting "as supplied."

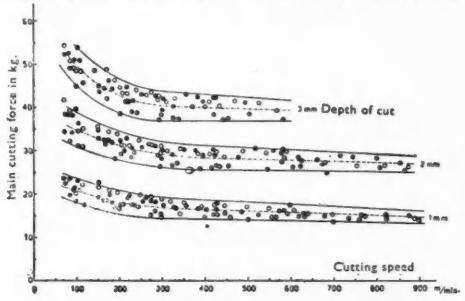
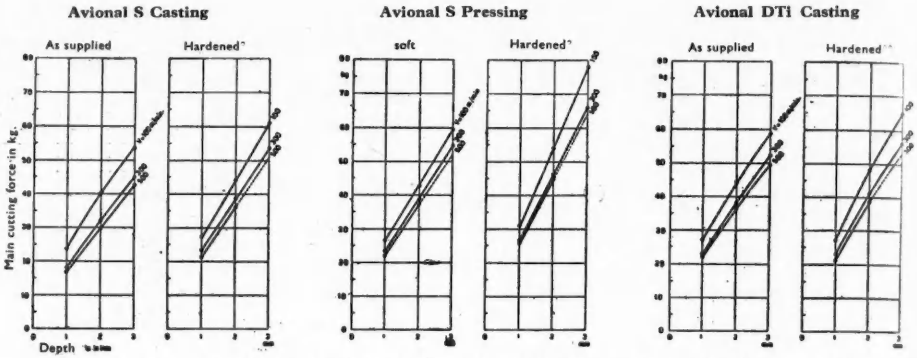


Fig. 7. **Machining test of Al-Si-Mg alloy Anticorodal casting "as supplied."**



° Annealed for 4 hours at 490-500 deg. C. and quenched. °° Annealed for 4 hours at 505-515 deg. C. and quenched.

Fig. 8. **Main cutting force v. depth of cut and cutting speed.**
Tool as per Fig. 4b. $\alpha = 8$ deg.; $\beta = 45$ deg.; $\kappa = 90$ deg. Rate of feed, 0.3 mm. per revolution.

These charts are exemplified by Figs. 5, 6 and 7, where it is seen that with increasing cutting speed the main cutting force tends asymptotically towards a limit value. This is a well-known phenomenon which is probably due to the decrease in friction between chip and tool face with increased cutting speed.

In Figs. 8a, b and c, referring to Avional, the mean values of the cutting forces for various cutting speeds are shown in relationship with the depth of cut. A characteristic feature of the cutting test is the coefficient of chip deformation which represents the ratio of the theoretical developed chip length to the actual (shorter) chip length. In general, the smaller this ratio the better is the quality of the machined surface. This deformation coefficient varies in straight proportion with the cross-sectional area of the chip. In Fig. 9 the values of the deformation coefficient found for the different materials investigated are plotted against the cutting speed. It is seen that particularly at low cutting speeds, the soft pure aluminium exhibits considerable chip deformation as compared to that registered with the aluminium alloys. Chip deformation and with it the deformation coefficient, is seen to decrease with higher cutting speeds.

Chip formation as such is conditioned by the deformability of the machined material, the lip angle of the tool, and the cutting speed. In the case of a discontinuous chip, the maximum deformability of the material is exceeded in the cutting process. A typical instance of this type of chip formation is the woolly chip produced by automatics, this being purposely produced by the choice of appropriate alloys. The production of a continuous chip is an indication that the maximum deformability of the material is not exceeded in the cutting process. A discontinuous chip will always be produced by a combination of very low cutting speed and a large lip angle of the tool. Since a continuous chip is synonymous with the highest quality of the machined surface, it follows that cutting should always be carried out at optimum speed.

Another characteristic phenomenon is the built-up edge consisting of material torn from the surface being machined and adhering to the working edge of the tool. During the cutting process this built-up edge is periodically torn off the tool edge, tearing off and re-forming, thus constituting a cyclic process which adversely affects surface quality. Formation of the built-up edge can be avoided by the choice of an adequately high cutting speed. It is also counteracted by efficient lubrication. The chips produced during the various tests conducted were consistently of the continuous type of varying length. In machining the cast materials it was observed that the chips tended to shorten with higher cutting speed, as will

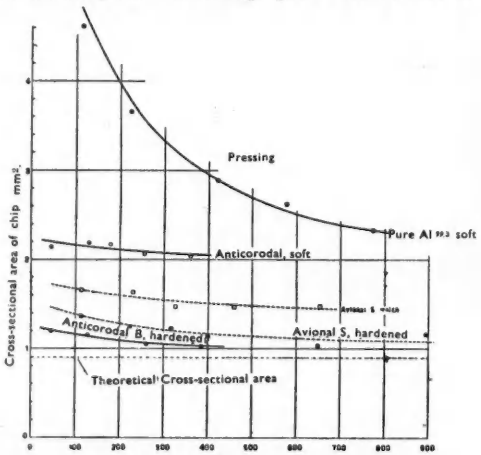


Fig. 9. **Chip deformation.**

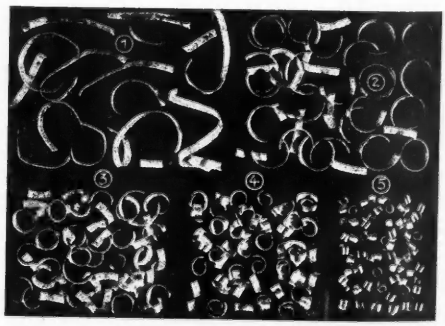


Fig. 10. **Chips obtained with machining of Avional-S casting "as supplied."** About ½ natural size.

No. 1	Cutting speed, m./min.		110
,, 2	,,	,,	220
,, 3	,,	,,	320
,, 4	,,	,,	440
,, 5	,,	,,	630

Fig. 11a

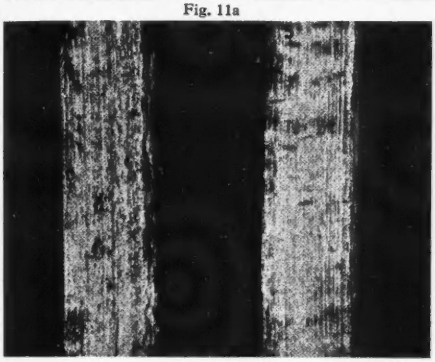


Fig. 11b

Fig. 11a, b. **Chips obtained with machining of Anticorodal casting "as supplied."** Magnif. 4.5 ×

No. 1	Cutting speed	110
,, 2	,,	230
,, 3	,,	320
,, 4	,,	450
,, 5	,,	630

be seen from Fig. 10. The photograph of the continuous chips obtained in the machining of cast Anticorodal shown in Figs. 11a and b clearly reveals a tendency towards the formation of smoother chips with higher cutting speed.

The mean cutting forces obtaining with the machining of the binary alloy Al + 5 per cent Cu at a cutting speed of 100 m. per minute were found to be larger than those recorded for either the pure aluminium or the Al + 10 per cent Cu alloy. This difference in the cutting force tends to disappear with the higher cutting speed, so that at 700 m. per minute cutting speed, the respective cutting forces of the three materials are practically identical. The following data were recorded with a depth of cut of 2 mm. and a feed rate of 0.3 mm.

	Cutting speed v (m./min.) 100	300	700
	Main cutting force, kg.		
Pure Al	34.4	31.2	29.6
Al + 5 per cent Cu as delivered ..	41.5	32.4	30.6
do. hardened ..	37.6	31.6	30.0
Al + 10 per cent Cu as delivered ..	34.8	30.6	29.4
do. hardened ..	34.4	31.2	29.6

The cutting forces observed in the machining of Anticorodal pressings were found to lie above those recorded for the cast material. Hardening was found to increase the cutting force on both the casting and the pressing. The cutting force values found are as follows :—

	Cutting speed v (m./min.) 100	300	700
	Main cutting force, kg.		
Casting :			
As supplied	34.0	29.0	27.5
Hardened	38.0	33.8	30.7
Pressing :			
Soft	40.0	34.2	31.2
Hardened	43.8	37.0	33.2

In the case of the Al-Cu-Mg alloy (Avional) it was noted that the cutting forces obtaining with the lower

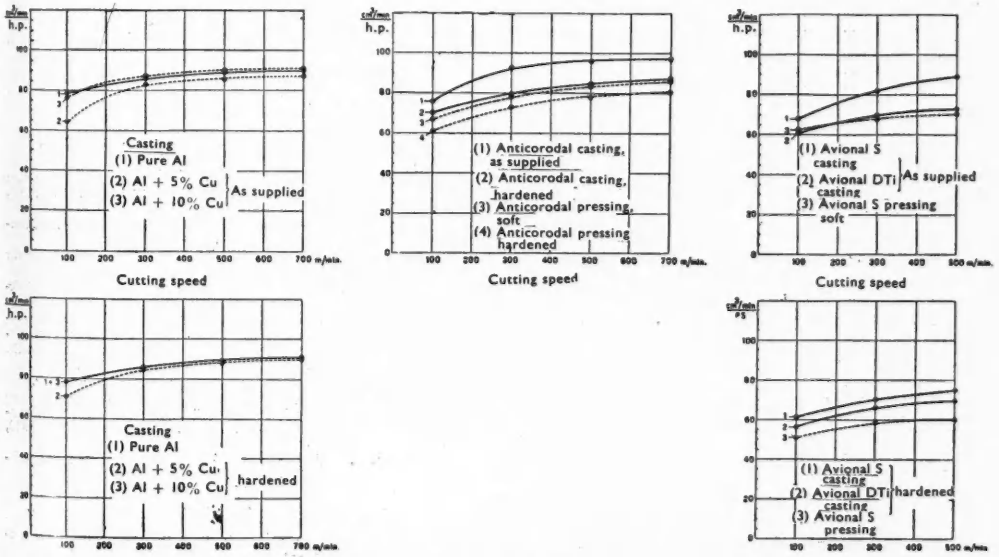


Fig. 12. **Specific output in cu. cm. per min., h.p.** versus **cutting speed.**

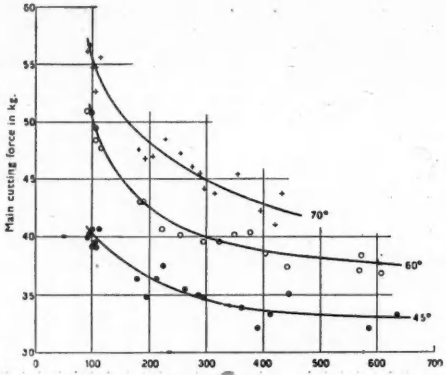


Fig. 13. **Influence of lip angle upon main cutting force and specific output.**

Material.—Anticorodal-B casting, machined with tool Fig. 4b without cutting fluid. Feed rate, 0.3 mm. Depth of cut, 2 mm $\alpha = 8$ deg ; $\beta = 4$ deg., 60 deg., 70 deg. ; $\kappa = 90$ deg.

alloyed cast material Avional DTi were considerably higher than with the Avional S cast material, particularly so when machining the former in "as supplied" condition. It has not yet been possible to ascertain whether or not the different cutting forces of the two materials are due to the greater structural fineness of the Avional DTi casting as compared to that of the Avional S casting. In the machining of the Avional S pressing greater cutting forces prevailed than with the cast variety of this material. This difference tends to spread with increased depth of cut. Hardening likewise leads to an increase in cutting force. For a depth of cut of 2 mm., the following data were noted.

	Cutting speed v (m./min.) 100	300	500
	Main cutting force, kg.		
Casting :			
Avional DTi :			
As supplied	43.6	38.0	36.6
Hardened	48.0	40.7	38.8
Avional S :			
As supplied	39.2	32.3	29.6
Hardened	43.8	38.4	36.2
Pressing :			
Avional S :			
As supplied	42.8	39.0	37.8
Hardened	53.2	46.2	45.9

The specific cutting forces in kg./sq. mm. obtaining with Anticorodal castings and also with Avional-S pressings were computed by dividing the main cutting force by the cross-sectional area of the chip, the latter being obtained from length and weight of the chip. It was found that for a constant rate of feed the specific cutting force decreased noticeably with increased depth of cut, especially so in the case of the hardened material. Since for the sake of high surface quality a minimum rate of feed should be combined with high cutting speed, optimum conditions will result if the cutting depth is chosen as large as possible. This will secure a maximum output per horse power supplied. The specific outputs in terms of cu. cm. per minute per h.p. ascertained in the machining of the various alloys are charted in Fig. 12.

As will be seen from Fig. 13, with all other conditions remaining equal, the magnitude of the main cutting force increases with the lip angle of the single-point tool. This chart refers to the cutting of a hardened Anticorodal casting, three different lip angles of 45, 60 and 75 deg. being used. By determining the corresponding specific outputs it was found that the output obtained with a lip angle of 70 deg. was 1.2 to 1.4 times that recorded with a lip angle of 45 deg. In continuous production this would correspond

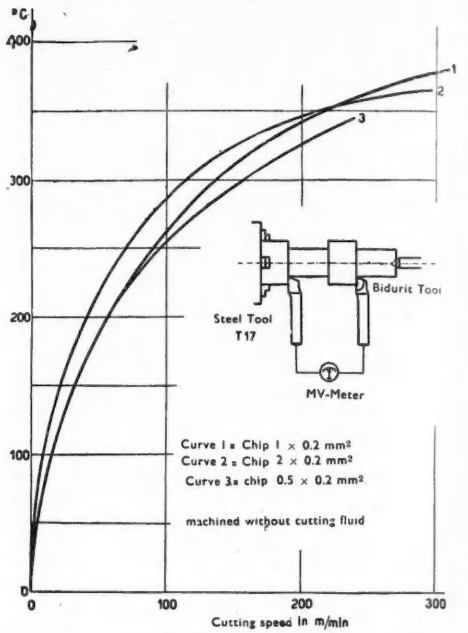


Fig. 14. **Determination of cutting edge temperature according to Gottwein-Reichel two-steel method.**
Material : Anticorodal pressing, hardened.

to a noticeable increase in power consumption.

In order to gain some information on the temperature prevailing at the cutting edge, an Anticorodal pressing in fully hardened condition was machined with the employment of the two-steel method, according to Gottwein-Reichel. In this method two single-point tools of different materials are used, the cutting points acting as a thermocouple, as indicated in Fig. 14. This chart was established on the basis of a constant feed rate of 0.2 mm., the depth of cut being chosen as 0.5, 1.0 and 2 mm. respectively. The cutting edge temperature is seen to increase sharply with the cutting speed, its trend being towards an upper temperature limit.

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WELDING CONDITIONS AND MECHANICAL PROPERTIES OF SPOT-WELDED ALUMINIUM SHEETS.

By R. IRMANN. (From *Schweizer Archiv*, Vol. 10, No. 6, June, 1944, pp. 177-181.)

IN judging the quality of a spot weld, the investigation of the mechanical properties of the weld ranks foremost. While additional information can be obtained from the visual appearance, weld quality generally depends upon the welding condition as well as upon the thickness of the welded sheet and its physical characteristics. Consideration of the process involved in the formation of a spot weld serves to supply a certain amount of pertinent information with regard to the welding conditions to be chosen. It is also helpful in forming a picture of the influences originating in the physical properties of the different aluminium alloys.

The welding current passing through the two sheets encounters the greatest electrical resistance (and thus creates the greatest heating effect) in the area of local contact between the sheets. By pickling or brushing the sheets prior to welding, the contact resistance between electrode and sheet is kept at a minimum; it also depends, however, upon the welding conditions. In the melting of the contact area of the sheets, the heat loss by conduction increases and the heating effect is greatly decreased because of the small electrical resistance of the sheets proper. High amperage is therefore required, but this must be combined with a brief welding period. These aspects are well illustrated by an examination of the influence of welding time and amperage upon the shear strength of Anticorodal-B sheet. Referring to Fig. 1, it is seen that the influence of welding time upon shear strength is very small in the case of thin material. The choice of the correct amperage is, however, of decisive influence upon the strength of the spot weld, this fact being evidenced by Fig. 2, which shows the interrelation of shear strength and amperage. The fact that with very high amperage the electrodes tend to stick to the sheet, however, prevents the application of very large currents. In every case the increase in

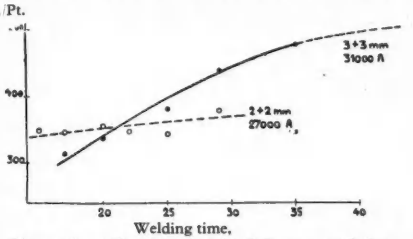


Fig. 1. **Shear strength of spot-welded 3 mm. and 2 mm. Anticorodal sheet as related to welding time.**

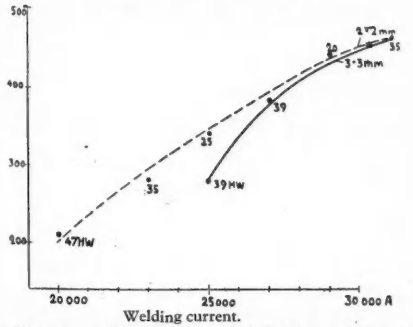


Fig. 2. **Shear strength of spot-welded 3 mm. and 2 mm. Anticorodal sheet as related to welding current intensity.**

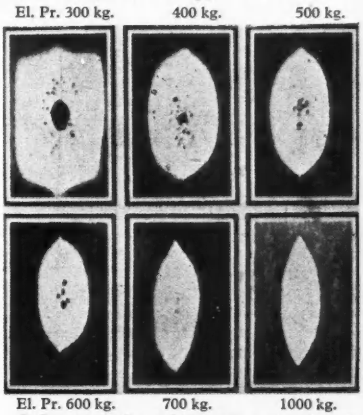


Fig. 3. **Influence of electrode pressure upon size of spot weld.** Magnif. 10 ×

spot weld strength is solely due to the increased area of the weld.

The size of the weld is dependent upon the electrode pressure, as the latter exerts a considerable influence upon the contact resistance between the sheets, and with it upon the extent of fusion produced. A variation in electrode pressure will therefore produce welds of different area, although welding time and current may be kept constant. This fact is evidenced by Fig. 3, showing spot welds produced with different electrode pressures ranging from 300 to 1,000 kg. The size of the weld is seen to increase with decreasing electrode pressure; but, as also demonstrated by Fig. 4, its porosity is seen to increase, the highest electrode pressure producing a weld of maximum density.

Attention must be drawn to the fact that the employment of high electrode pressure, to be successful, requires the use of slightly spherical electrode tips in order to avoid undesirable indentation of the sheet. High electrode pressure is of especial influence upon the scatter of strength values, which may be considerable with adverse welding conditions. The influence of electrode pressure in this respect is illustrated by Fig. 5, which shows the decrease in strength value

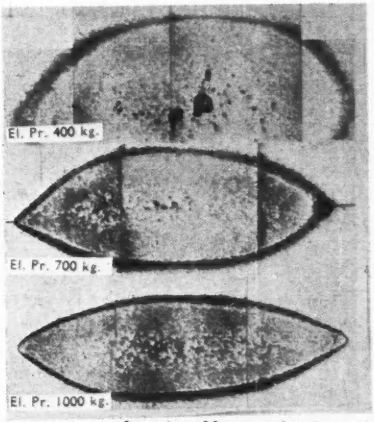


Fig. 4. **Appearance of spot welds as related to electrode pressure. 3 mm. Anticorodal sheet.** Magnif. 30 ×

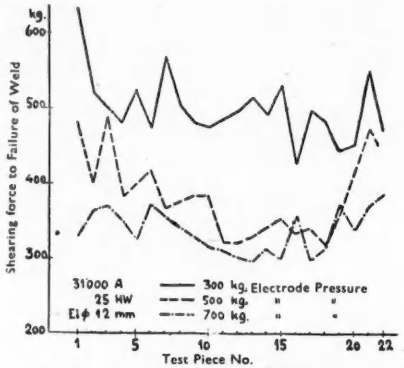


Fig. 5. **Influence of electrode pressure upon scatter of shear strength values.** (Spot-welded 3 mm. Anticorodal-B sheet.)

scatter with higher electrode pressures. This observation also permits the gauging of the favourable effect to be expected from operation with a controlled variable electrode pressure, characterised by a lowering of the high initial pressure during the time of the passage of the welding current.

The inter-relationship which exists in spot welding between the scatter of the shear strength values and the surface condition is still awaiting clarification. So far, the author's investigations solely permit the conclusion that scatter increases with the thickness of the material. With thick materials the fusion zone shows a greater tendency to reveal such irregularities as porosities and fissures, the formation of which is favoured by rapid cooling and high temperature gradient respectively. In this respect also, the employment of controlled variable electrode pressure appears to be advantageous, this process causing compression of the weld subsequent to cessation of the welding current. In addition to closing porosities and fissures, a slight working of the metal is thereby effected.

Fundamentally it would be desirable to replace the fusion welding effect occurring in spot welding by a diffusion welding process similar to that obtained with the hammer welding of aluminium. In addition to the employment of a higher temperature, the process of hammer welding requires the occurrence of a sufficiently high pressure combined with a sliding movement, the latter destroying the oxide film which forms an obstacle to diffusion. It would appear that the recently proposed employment of vibrating electrodes in the spot welding process is aiming in this direction.

As will be seen in Fig. 6, with spot welding proper, complete destruction of the oxide film is effected solely in the fusion zone, while oxide films still remain plainly visible in the pasty transition zone, its temperature lying within the range delimited by solidus and liquidus temperatures respectively. The employment of diffusion welding for joining different metals such as copper and aluminium may be considered the only means of effecting a seam free from the harmful weld embrittlement experienced with fusion welding. It is interesting to note that it has been suggested to apply a system of combined electrode pressure and current control to the heat treatment of the finished seam.

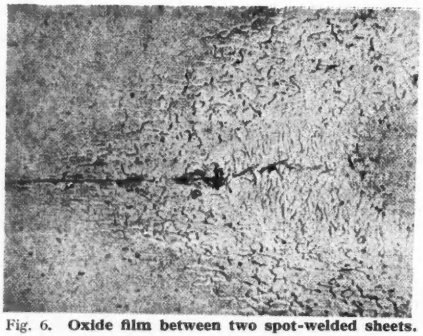


Fig. 6. **Oxide film between two spot-welded sheets.**

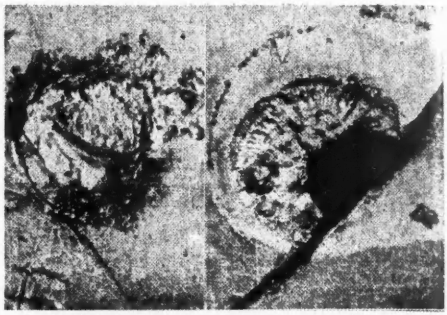


Fig. 7. **Fatigue fracture of spot weld in Anticorodal sheet.**

All this goes to show that the quality of the weld is dependent upon the state of the structure. Generally, valid specifications for the welding of a given material cannot therefore be established as yet.

The fatigue failure of spot weld has its inception at the point of maximum stress—that is, at the periphery of the seam where stress values five times the average stress are found to occur. Two spot weld fatigue failures are illustrated in Fig. 7, representing views of the fractures on the contacting side of the sheet. The line of fatigue failure is seen to follow the periphery of the seam and then to extend, in part, into the fusion zone of the seam proper. It is interesting to note that unlike the static strength, the fatigue strength does not increase with the diameter of the seam. This is explained by the fact that fatigue failure commences at the point of greatest stress concentration coincident with the periphery of the seam. The diameter of the spot weld itself is therefore of little importance.

In a riveted joint the line of fatigue fracture follows a course similar to that described above—that is, the line of fracture follows the periphery of the head of the rivet. While riveted and spot-welded joints are comparable from the aspect of static shear strength, no such comparison is permissible with respect to the fatigue strengths of these two types of joint.

Referring to Fig. 8, both a riveted joint and an autogeneous weld joining 1 mm. Anticorodal sheet yield a fatigue strength of 7 kg. per sq. mm., as compared to a fatigue strength of 9 kg. per sq. mm. of the sheet proper; the spot-welded joint shows a fatigue strength of only 5 kg. per sq. mm. Spot-welded joints should not be subjected to cyclic bending stress.

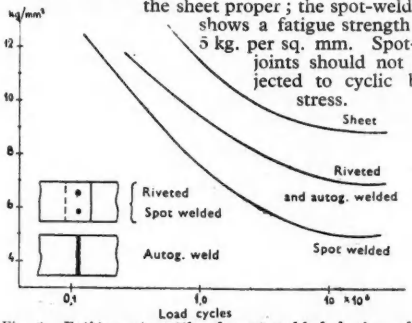


Fig. 8. **Fatigue strength of spot-welded Anticorodal sheet (1 mm. thick) as compared to riveted, autog.-welded and original sheet.**

CONNECTING ROD ACCELERATION.

By Dr. Ing. Ernst Frey, Altena. (From *ATZ—Automobiltechnische Zeitschrift*, Vol. 46, Nos. 7/8, April 25th, 1943, pp. 178-183).

ACCELERATION IN THE DEAD CENTRE POSITIONS.

The imaginary centre of the connecting rod big end moving along the crank circle has a constant centripetal acceleration $r\omega^2$. The imaginary centre of the little end of the connecting rod moves up and down along a straight line and has the same acceleration as the piston. Any other point on the connecting rod describes a path similar to an ellipse. It is well known that the piston acceleration is,

$$p = r\omega^2 (1 + \lambda), \quad \text{for } \alpha = 0^\circ \text{ (O.D.C)}$$

and $$p = r\omega^2 (1 - \lambda), \quad \text{for } \alpha = 180^\circ \text{ (I.D.C.)}$$

In the following, a method is given by means of which these two equations can be found without the usual double integration of the equations for piston displacement, and which gives also the acceleration of any point of the connecting rod.

If the connecting rod is considered to be moved along the crank circle, parallel to its original position in the O.D.C. (Fig. 1), then every point on the connecting rod, including the imaginary centres, has a downward acceleration of $r\omega^2$. In reality, the connecting rod is also rotating around the crank pin with an angular velocity equal to r/l times the crank velocity and in the opposite direction. Owing to this rotation, the centre of the little end has an additional downward acceleration,

$$l \left(\frac{r}{l} \omega \right)^2 = \lambda r\omega^2$$

The total acceleration is, therefore, $p = r\omega^2 (1 + \lambda)$.

In the O.D.C. position, any point along the centre line of the connecting rod, at a distance e from the big-end centre, has a resultant acceleration,

$$p_e = r\omega^2 + e \left(\frac{r}{l} \omega \right)^2 = r\omega^2 \left(1 + \frac{e}{l} \lambda \right) \quad .. \quad (1)$$

$\alpha = 0^\circ$

In the I.D.C. position the acceleration due to rotation of the connecting rod is directed downward while the centripetal acceleration is in the upward direction. Thus,

$$p_e = -r\omega^2 + e \left(\frac{r}{l} \omega \right)^2 = -r\omega^2 \left(1 - \frac{e}{l} \lambda \right) \quad .. \quad (2)$$

$\alpha = 180^\circ$

The acceleration of any point on the centre line is, therefore, as shown in Figs. 2 and 3. In the O.D.C. the acceleration of a point on the connecting rod axis increases uniformly from $r\omega^2$ (big end centre), to $r\omega^2 (1 + \lambda)$ (little end centre), while in the I.D.C. it decreases from $r\omega^2$ to $r\omega^2 (1 - \lambda)$. It can be seen that in the I.D.C. positions the point situated at a distance l/λ above the big end centre, and in the O.D.C. position the point situated the same distance below the big end centre have both zero acceleration. At distances greater than l/λ the acceleration in the D.C. positions is even reversed.

The acceleration of points situated outside the centre line is determined in the same way. As Fig. 4 shows, the accelerations p_{a1} and p_{a2} of two points A_1 and A_2, symmetrically situated at the two sides of the centre line, are greater than that of the point A, but their vectorial sum $p_{a1} \rightarrow p_{a2} = 2\, p_a$. Thus, all points on the cross section normal to the centre line may be treated in the same way when the inertia forces are calculated. For the whole of the cross section the acceleration is as indicated in Figs. 2 and 3. In order to determine the inertia forces in the D.C. positions, the connecting rod is sub-divided into 10–15 elements Δm, each of them is multiplied by the corresponding acceleration, and their sum total determined. The total inertia force is thus,

$$\Sigma\, r\omega^2 \left(1 \pm \frac{e}{l} \lambda \right) \Delta m$$

If the center of gravity of the connecting rod is at a distance $e = \xi$ from the crankpin centre, and if $\Sigma \Delta m = m$ = mass of the connecting rod, then

$$P_0^{180^\circ} = mr\omega^2 \left(1 \pm \frac{\xi}{l} \lambda \right)$$

$$= m\, \omega^2 [r \pm \lambda^2 \xi] \quad .. \quad .. \quad (3)$$

If the mass of the connecting rod is divided into two parts and considered to be concentrated at the centre of the crankpin (m_r) and at the centre of the gudgeon pin (m_s) respectively, so that their centre of gravity coincides with the actual centre of gravity of the connecting rod, then

$$P_0^{180^\circ} = m_r\, r\, \omega^2 + m_s\, r\, \omega^2 (1 \pm \lambda) \quad .. \quad (3a)$$

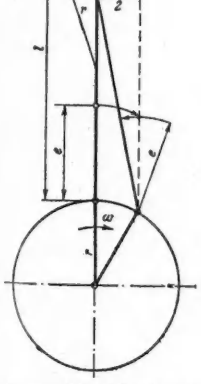


Fig. 1. **Resolving the connecting rod movement into a component parallel to the connecting rod axis (1) and a rotation (2) around the crank pin.**

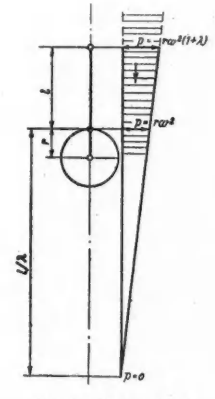


Fig. 2. **Diagram of accelerations at O.D.C.**

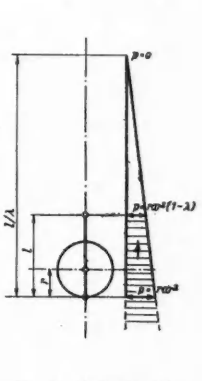


Fig. 3. **Diagram of accelerations at I.D.C.**

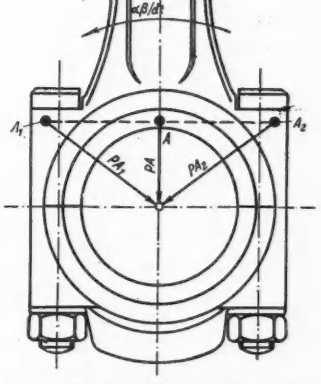


Fig. 4. **The vectorial sum of the centripetal accelerations of A_1 and A_2 is equal to twice the centripetal acceleration of A.**

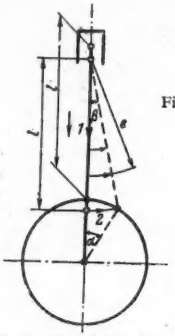


Fig. 5

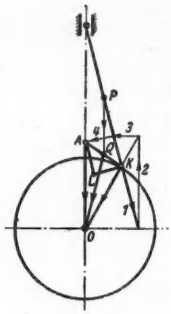


Fig. 6

Fig. 5. **Resolving the connecting rod movement into an axial component and a rotation around the gudgeon pin centre.**
Fig. 6. $\overline{QO}$ = **acceleration of** P. $\overline{AO}$ = **piston acceleration.**

For the usual type of connecting rods the total inertia force becomes :

	O.D.C.	I.D.C.
for $\lambda = 1.5$	$1.07\ mr\omega^2$	$0.93\ mr\omega^2$
for $\lambda = 1.4$	$1.08\ mr\omega^2$	$0.92\ mr\omega^2$
for $\lambda = 1.36$	$1.09\ mr\omega^2$	$0.91\ mr\omega^2$

ACCELERATION AT ANY ARBITRARY CRANK POSITION.

The acceleration of a point on the connecting rod at any arbitrary crank position may be found, as before, by adding the acceleration due to an axial movement of the connecting rod parallel to the piston, and the acceleration due to rotation around the gudgeon pin, as shown in Fig. 5.

(1) The downward (piston) acceleration is,

$$\frac{d^2s}{dt_2} = r\omega^2\left(\frac{\cos(\alpha+\beta)}{\cos\beta} + \lambda\frac{\cos^2\alpha}{\cos^3\beta}\right);$$

this is the same for every crank position.

(2) The tangential acceleration due to rotation with the gudgeon pin as centre, is

$$e\,\frac{d^2\beta}{dt^2}$$

where e is the distance of the point considered from the centre of the little end. By double differentiation of $\beta = \sin^{-1}\lambda\sin\alpha$.

$$\frac{d^2\beta}{dt^2} = \lambda\omega^2\left(\frac{\lambda\cos^2\alpha\sin\beta}{\cos^3\beta} - \frac{\sin\alpha}{\cos\beta}\right) =$$

$$= \lambda\omega^2\left[\frac{(\lambda^2-1)\sin\alpha}{(1-\lambda^2\sin^2\alpha)^{1.5}}\right] = \lambda\omega^2\left[\frac{(\lambda^2-1)\sin\alpha}{\cos^3\beta}\right]$$

For $\alpha = 0°$ *or* $180°$, $\dfrac{d^2\beta}{dt^2} = 0$

For $\alpha = 90°$ or $270°$, the angular acceleration is a maximum, and the acceleration of the centre of the big end, perpendicular to the connecting rod axis is,

$$l\,\frac{d^2\beta}{dt^2}_{\text{max.}} = \frac{r\omega^2}{\cos\beta} = \frac{r\omega^2}{\sqrt{1-\lambda^2}}$$

(3) The centripetal acceleration, corresponding to the relative rotation of the connecting rod is,

$$e\left(\frac{d\beta}{dt}\right)^2 = e\left[\lambda\omega\frac{\cos\alpha}{\cos\beta}\right]^2$$

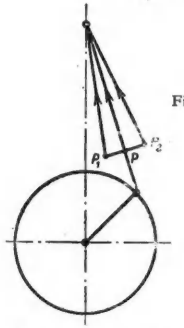


Fig. 9

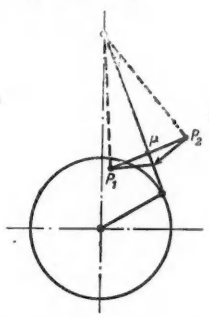


Fig. 10

Fig. 9. **The vectorial sum of the centripetal accelerations of** P_1 **and** P_2 **is equal to twice the centripetal acceleration of** P.
Fig. 10. **The vectorial sum of the tangential accelerations of** P_1 **and** P_2 **is twice the tangential acceleration of point** P.

A mathematical computation of these accelerations for a large number of crank positions is very involved; a graphical method is, however, available. According to Fig. 6, the lines 1, 2, 3, 4 are drawn (*ATZ*, 1940, No. 15, p. 381) and the piston acceleration is then found to be $\overline{AO} = \dfrac{d^2s}{dt^2}$. Since the direction and magnitude of the resultant acceleration of the point K is known to be $p_k = \overline{KO} = r\omega^2$, the side $\overline{KA}$ of the triangle $AOKA$ must be the resultant of the tangential acceleration ($\overline{KL}$) and of the centripetal acceleration ($\overline{LA}$) of the point K. Considering that both accelerations, $e\left(\dfrac{d^2\beta}{dt^2}\right)$ and $e\left(\dfrac{d\beta}{dt}\right)^2$, of any point on the centre line are proportional to the distance e, and have the same directions as the component accelerations of the point K, then the resultant acceleration of a point P is given by $\overline{QO}$ of the triangle $QAOQ$, which is obtained by drawing the lines $\overline{PQ}\,//\,\overline{AO}$. The scale is given by $\overline{KO} = r\omega^2$. The acceleration of a number of points is shown in Figs. 7 and 8. Fig. 9 shows that the resultant centripetal acceleration of two points, P_1 and P_2, symmetrically situated at the two sides of the centre line, is equal to twice the corresponding acceleration of the point P. Similar is the case for the tangential acceleration (Fig. 10), but the resultant acceleration of the points P_1 and P_2 does not pass through the point P. Strictly speaking, therefore, the acceleration of points outside the centre line should be determined individually. This may be done graphically. If only the sum total of the inertia forces is required, the acceleration of a section of the connecting rod can be assumed to be approximately twice that of a point on the centre

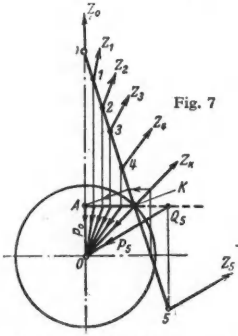


Fig. 7

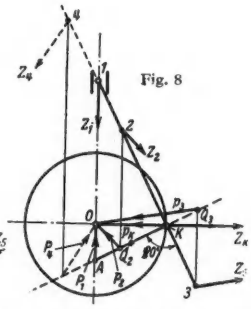


Fig. 8

Fig. 7. **Accelerations** p **and inertia forces** z **of arbitrary points on the centre line of the connecting rod.**
Fig. 8. **Accelerations** p **and inertia forces** z **at** $\alpha = 90$ **deg.** $\left(\dfrac{d\beta}{dt} = O\right)$

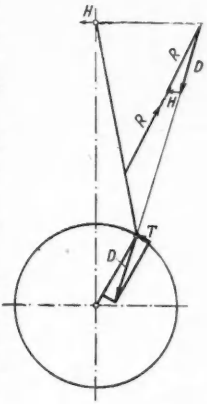


Fig. 11. **Resolving the inertia force R into forces H and D.**

line and determined according to Figs. 7 and 8. The connecting rod is subdivided into a convenient number of masses Δm, and each is multiplied by the corresponding acceleration. The values so found are drawn to scale, outward from the centre line. This gives a series of forces, z, e.g., Figs. 7 and 8. The resultant of these forces is found by drawing a force polygon (Hütte I, 26th edition, p. 213). Finally, the forces acting at the connecting rod centres are found as shown in Fig. 11. The force at the gudgeon pin, H, acts in a horizontal direction (perpendicular to the axis of the crank shaft). The forces, R, D and H meet in one point and, therefore, the triangle of forces can be drawn and their magnitude determined. The force, D, acting at the crankpin is composed of a radial force and a tangential force, T. The force polygon need not be drawn for R; by means of the force polygon the forces acting at the connecting rod centres can be determined also from the individual forces, $z_1 - z_n$. (Hütte I, 26th edition, p. 326, Fig. 54).

A still simpler method is illustrated in Fig. 12. Each force $z_0 - z_6$ is resolved into a gudgeon pin reaction and a crankpin reaction, and these are compounded to give D and H. Finally D is resolved into a turning moment T and a radial force.

This method (Fig. 12) can be used if the mass, the centre of gravity and the moment of inertia of the connecting rod are not known; it requires, however, a number of drawings (at least 11) of the type shown in Fig. 12. Much work can be saved if the mass, the centre of gravity, the static moment and the moment of inertia of the connecting rod are first determined. The simplest method is by experiment with the actual con-

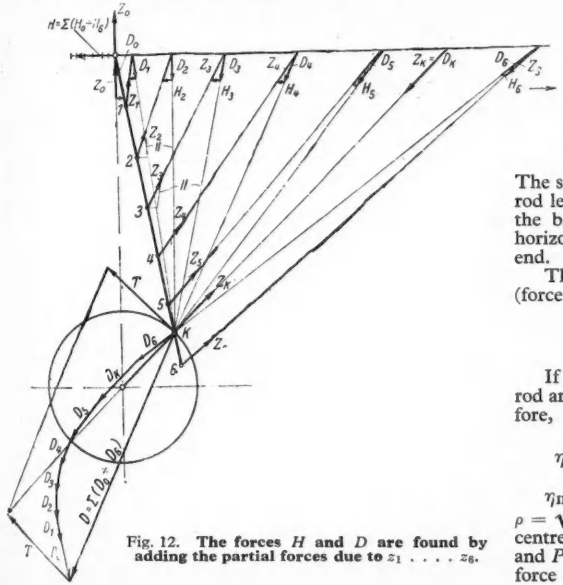
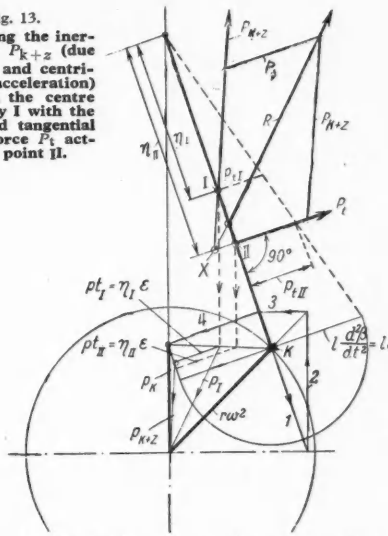


Fig. 12. **The forces H and D are found by adding the partial forces due to $z_1 \ldots z_6$.**

Fig. 13. **Combining the inertia force P_{k+z} (due to piston and centripetal acceleration) acting in the centre of gravity I with the calculated tangential inertia force P_t acting in point II.**



necting rod. Then for each crank position, (a) the inertia force due to the resultant piston and centripetal accelerations, p_x and p_z, and (b) the inertia force due to the tangential acceleration of the connecting rod are determined. The resultant of these is found by drawing the force polygon.

(a) The acceleration diagram (Fig. 13) gives the resultant $p_k \rightarrow p_z$ for the centre of gravity, I, of the connecting rod $p_k \rightarrow p_z$ multiplied by the total mass m gives the inertia force p_{k+z} acting at the centre of gravity, at a distance η_I from the gudgeon pin centre.

(b) The inertia force due to rotation, of an element dm at a distance e from the gudgeon pin centre is,

$$dP_t = e \frac{d^2\beta}{dt^2} dm = e\,\epsilon\,dm,$$

and the total force due to rotation is

$$P_t = \epsilon \int e\,dm = \frac{e}{g} \int_{e=o}^{e=l} e\,dG,$$

$$\text{or } P_t = \frac{e}{g} M_{st}\ (\text{kg.}) \quad .. \quad .. \quad ..$$

The static moment M_{st} is the product of the connecting rod length (l) multiplied by the vertical force acting on the big end centre when the connecting rod is in a horizontal position and free to rotate around the little end.

The distance of point II from the little end centre (force P_t acts at point II) is given by

$$\eta_{II} = \frac{M_d}{P_t}$$

If $\mathcal{J}$ denotes the moment of inertia of the connecting rod around the little end centre, then $M_d = \mathcal{J}\epsilon$. Therefore,

$$\eta_{II} = \frac{\mathcal{J}\epsilon}{P_t} = \mathcal{J}\epsilon \frac{g}{\epsilon M_{st}} = \frac{gJ}{M_{st}}\ (\text{meter}) \quad .. \quad (5)$$

η_{II} is neither equal to η_I nor to the radius of gyration $\rho = \sqrt{\mathcal{J}/m}$. P_t is marked off at a distance η_{II} from the centre line, at 90 deg. to it. The resultant R of P_{k+z} and P_t is found by completing the force diagram. The force S acting at the crankpin centre passes through the

point of intersection of R and a horizontal line drawn through the little end centre.

The method illustrated in Fig. 13 can be further simplified. It is sufficient to determine the point of intersection x, of P_{k+z} and P_t. The inertia force R passes through this point and its magnitude is given by $R = p_{Im}$. Its direction is the same as that of the acceleration p_I, of the centre of gravity. Fig. 14 shows a graphical method of determining R. After the centre of gravity I, and the centre of gravity of the inertia forces II are found, the lines 1–11 are constructed, the line 12 drawn parallel to 10, and 13 perpendicular to the connecting rod axis, and finally R parallel to line 11.

This method can be made very simple by a suitable choice of the scale. The crank radius is usually drawn, $r = \frac{mr\omega^2}{F}$, where 1 cm. = 1 or 2 kg./cm.² piston area.

From R only the reactions at the connecting rod centres can be determined. If the stresses in the connecting rod are to be investigated, the latter must be subdivided into a number of elements, as before.

It would be of general interest to determine the tangential inertia forces acting on the crankpin for all crankpin positions between $\alpha = 0$ deg. and $\alpha = 180$ deg. This would show whether it is justified to add a part of the connecting rod weight to the piston, as is usually done in practice when the inertia forces are calculated.

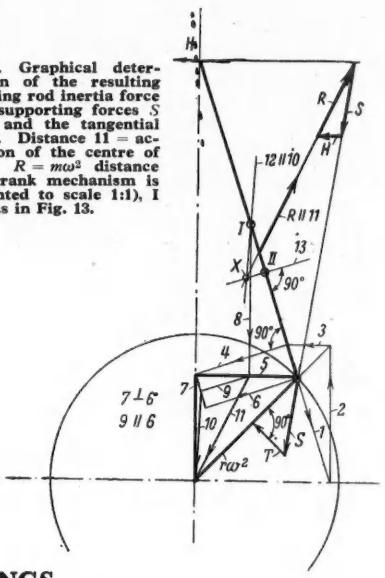


Fig. 14. **Graphical determination of the resulting connecting rod inertia force R, the supporting forces S and H and the tangential force T. Distance 11 = acceleration of the centre of gravity. $R = m\omega^2$ distance 11 (if crank mechanism is represented to scale 1:1), I and II as in Fig. 13.**

PLASTIC BEARINGS

By E. HEIDEBROEK, V.D.I., Dresden, and H. ZICKEL, V.D.I., Troisdorf. (From *Kunststoffe*, Vol. 33, No. 10, October, 1943, pp. 243-249).

EXPERIENCE with plastic bearings during the last few years has been generally satisfactory, but there were also cases of failure. One reason for failure was due to lack of uniformity of the plastic materials used for bearings, and in that respect the type of filler is of particular importance. Swelling of the material is another variable factor influencing the results as well as the structure and the nature of the bearing surface which affects lubrication, particularly under boundary lubrication conditions.

For a practical evaluation of the influencing factors, it was necessary to carry out comparative tests with various types of materials under carefully controlled test conditions. The result of these tests is discussed in the following.

TABLE I.

	Type of Resin	Resin Content
Pressed T–2 Bushes with Cellulose Shaving Filler.		
L 1	Phenol—Novolak	42 %
L 3	Phenol—Resole	42 %
L 5	Kresol—Novolak	42 %
L 7	Kresol—Resole	42 %
L 24	Phenol—Kresole—Resin	42 %
L 25	Phenol—Kresole—Resin	42 %
L 26	Phenol—Kresole—Resin	42 %
L 27	Phenol—Kresole—Resin	35 %
Pressed T—2 Bushes with Mixed Fabric—Shred Filler		
L 9	Phenol—Novolak	42 %
L 11	Phenol—Resole	42 %
L 13	Kresole—Novolak	42 %
L 15	Kresole—Resole	42 %
L 28	Phenol—Kresole—Resin	42 %
L 29	Phenol—Kresole—Resin	42 %
L 30	Phenol—Kresole—Resin	42 %
L 31	Phenol—Kresole—Resin	35 %

TABLE II.

THERMOPLASTIC MATERIALS WITHOUT FILLER.

	Type of Material
L 38	Hard Mipolam
L 39	Vinidur

	Pressed T—2 Bushes with Type of Resin	Resin Content
L 35	Phenol—Kresole—Resin	42 %
L 36	Phenol—Kresole—Resin	42 %
L 37	Phenol—Kresole—Resin	42 %

Tables I and II show the materials investigated. The hardness of the materials was measured with a Shopper-type apparatus, with 50 kg. load, 10 mins. loading time, and a 5 mm. dia. ball indentor. The results of these tests are shown in Fig. 1, but it should be born in mind that since the indentor penetrates beneath the surface of the material into the filler, no conclusions could be drawn regarding surface hardness of the material.

The bushes were supplied by Messrs. Dynamit A.G., and on some thermoplastic bushes a slight ovality was observed.

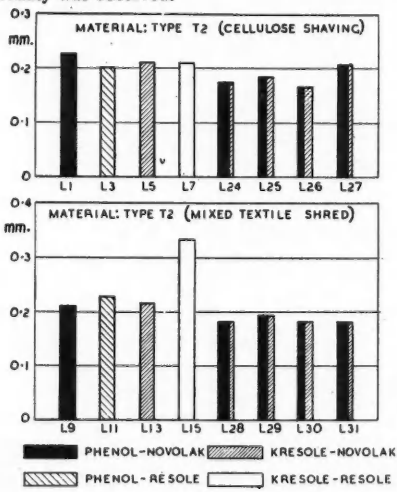


Fig. 1.

A standrard oil (*BC*8) and two types of synthetic oils (S_4 and S_2) were used in order to gain some information also on the influence of lubricants on the bearing properties. The bearings were tested under static loading conditions.

TEST APPARATUS AND METHOD OF TESTING

A schematic arrangement of the test apparatus developed by the Technical College of Dresden, to test bearings up to 60.25 mm. dia. and 40 mm. width, is shown in Fig. 2. The oil inlet temperature and pressure were kept constant at 30 deg. C. and 1 atmos. respectively throughout the test. The oil film temperature was also measured, approximately at the point of maximum load by means of a thermocouple fixed in the surface of the bearing. Provision was also made to measure the oil flow.

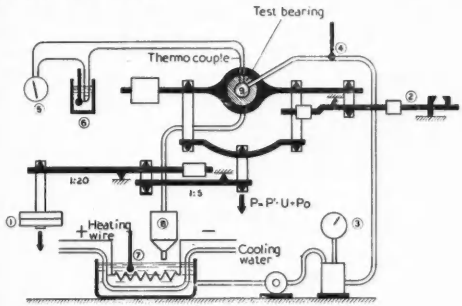


Fig. 2.
Measurements.
(1) Load (kg.) (2) frictional torque (mkg.) (3) oil pressure (atmos.) (4) oil inlet temperature (deg. C.) (5) galvenometer (6) cold juntion temperature (deg. C.) (7) temperature in oil tank (deg. C.) (8) rate of oil flow (cu. cm. /sec.) (9) shaft speed (r.p.m.).

The friction torque, M_{dr}, was measured by hanging weights on the bearing shell and thus balancing the torque from which the value of the coefficient, μ, could be calculated by means of the relationship,

$$\mu = \frac{M_{dr}}{P \cdot r}$$

where r is the journal radius and P the load. The value of μ depends upon the surface roughness of the bearing, the viscosity of oil, the wetting property of the surface of oil, etc., and therefore, represents an aggregate value.

The bearing clearance was in all cases 0.25 mm., and the same shaft could be used repeatedly since the tests were stopped before seizure occurred. The values of μ were plotted against speed, and the shape of the curves clearly indicates the point of change over from hydrodynamic lubrication to boundary lubrication (Fig. 3). In this respect greatly varying characteristics were observed with the different materials used.

COMPARISON OF MATERIALS.

It is known from the hydrodynamic theory of lubrication that the clearance between journal and bearing depends upon speed and load. If the speed falls below a certain minimum value, the oil film breaks down and metal to metal contact is established between journal and bearing. This condition can be

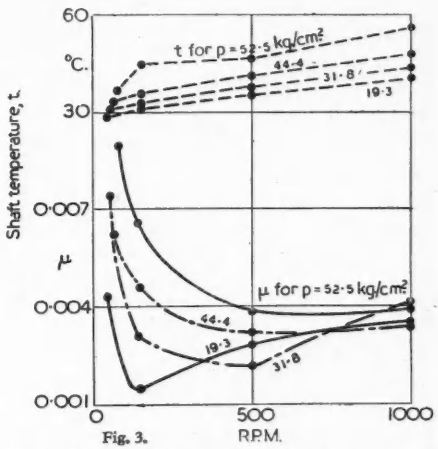


Fig. 3.

noted by the unstable operation of the test apparatus, e.g., flickering of the ammeter needle registering the input to the main driving motor, or oil pressure variation. Alternatively, the speed can be kept constant at a reasonably high value and the load increased until the bearing temperature shows a marked tendency to rise rapidly. With plastic bearings particularly there is a rather strict limit to the maximum permissible temperature which is about 80 deg. C. for T-2 type bearings, so that by observing the shape of the temperature-load curve a limiting load can be noted.

As a quantitative basis of comparison of plastic bearing materials, therefore, these two characteristic values, namely speed (n_{min}) and maximum load (p_{max}) were made use of, both of which are indenepent of μ and represent practical limitations.

FRICTION, HEAT, SWELLING, RATE OF OIL FLOW.

At high journal speeds, the criterion of satisfactory operation is given by the heat balance, that is the heat generated by friction and the heat carried away by the oil flowing through the machine, by radiation and by conviction. In the test machine practically the total heat generated had to be carried away by the oil, since the heat conductivity of plastic materials is very low,

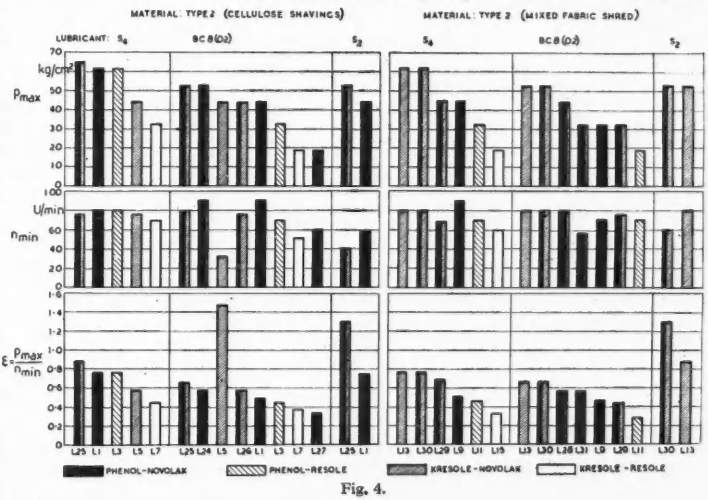


Fig. 4.

and the arrangement of the test apparatus was such that very little heat could be conducted away through the journal or by radiation. For this case, therefore,

$$\frac{P \cdot \mu \cdot v}{427} \text{ kcal./sec.} = Ar = Q\,(t_2 - t_1)\,c_0 \quad \text{and}$$

$$Q = \frac{Ar}{(t_2 - t_1)\,c_0} \text{ kg./sec.}$$

where Q is the quantity of oil pressing through the machine per second, t_2 and t_1 respectively the outlet and inlet temperature of oil, and c_0 the specific heat of oil.

In order to eliminate the effect of swelling, which varies for the different types of material, all bushes were designed with a clearance of 0.25 mm. on 60 mm. diam. This clearance was ample so that differences in swelling did not introduce an appreciable diminution of oil flow.

An interesting feature of plastic bearings is that, clearance for clearance they permit a larger quantity of oil flow than metallic bearings. This appears to be anomalous, but in all probability it is due to the surface effect of plastic materials which results in an orientation of the oil molecules. In so far as it helps to keep the temperature down, it is an advantageous feature of plastic bearings.

TEST RESULTS.

A very large number of tests have been carried out, and it is, therefore, only possible to give a summary of the results. A typical example of performance of *T-2* type bearings is shown in Fig. 3. It can be seen that as the load increases, the condition of boundary lubrication is moved to the right hand side, i.e., to higher speed. The temperature also increases with speed and reaches 60 deg. C. with a load of 52.5 kg./cm² at 1000 r.p.m. The limiting load, as can be seen from Fig. 3, for this type of plastic is 52.5. kg./cm². and co-ordinated to this load is a min. speed (n_{min}) of 80 r.p.m. In a similar manner values of (p_{max}) and (n_{min}) were determined for the other materials. Since the bearing capacity is directly proportional to load (p_{max}) and indirectly to the speed (n_{min}) this can be expressed conveniently as

$$\epsilon = \frac{p_{max}}{n_{min}}$$

which is a useful parameter for making comparisons. (It is only a coincidence that the dimension of ϵ are $\frac{\text{kg./sec.}}{cm.^2}$ which is identical with that of viscosity).

Fig. 4 gives the tabulated results. The three columns refer to tests with synthetic oil (S_4), standard oil *BC8*, and a synthetic oil of high viscosity S_2.

Fig. 4 shows that the performance varies for the same type of resin with different filler materials, and to a lesser extent with the type of lubricant. This means that not any type of filler can be used in conjunction with a certain type of resin.

The bearing properties are independent of hardness, since Fig. 1 shows no appreciable differences in hardness numbers, as compared with the results shown in Fig. 4. Similarly the resin content (Table I) is of little importance.

It is interesting to note that of the special type of plastics, fillers of jute fibres did not show up well, contrary to expectations.

THERMOPLASTIC BEARING MATERIALS.

On account of softening at relatively low temperatures, thermoplastic materials were not expected to be of much use for bearing applications. As can be seen from Fig. 5, however, Mipolam has shown good results with loads of about 27 kg./cm²., and the high rate of oil flow through the bearing (Fig. 6) indicates very low bearing temperatures. These tests have proved that there is a useful field for thermoplastic bearings—operating at high speed and comparatively low loads.

Fig. 7 shows a graphical representation of the limiting load and speed. In this case the upper limit of velocity was arbitrarily chosen as 2000 r.p.m. or 6.28 m. sec., but since the temperature rise with increasing speed is rather flat, the diagram could be extended to above 2000 r.p.m. provided that the value of pv is not exceeded. As the rate of oil flow increases rapidly with increasing speed, it can be safely assumed that sufficient cooling takes place also at higher speeds.

WATER LUBRICATION.

Experiments were also carried out with water cooling. As can be seen from Fig. 8, fully hydrodynamic conditions of lubrication were attained at a speed of $r = 4.7$ m. sec., and the value of p_{max} does not exceed 20 kg./cm². Thus with water lubrication stable conditions are reached only at relatively high speed, and until this condition is reached the danger of seizure is prevalent. On the other hand, owing to the lower viscosity of water as compared with oil, the rate of flow of water is much greater and combined with the high specific heat of water the temperature conditions are very satisfactory. Experience in rolling mills confirms these results. Although the load is rather high (200

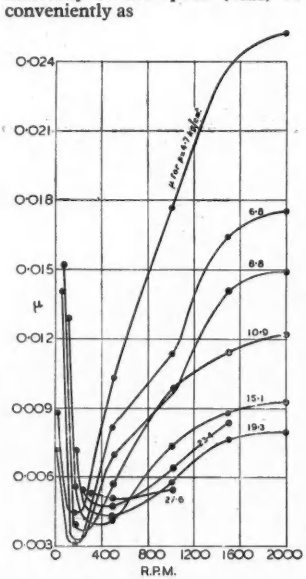


Fig. 5.

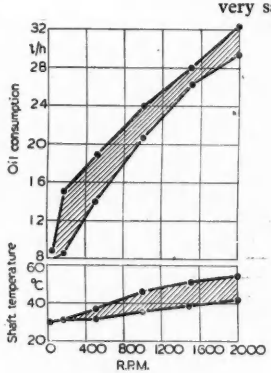


Fig. 6.

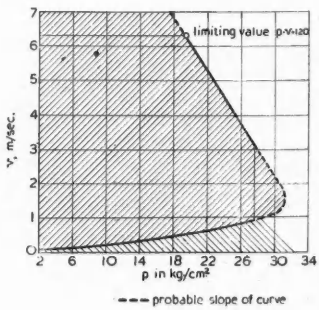


Fig. 7.

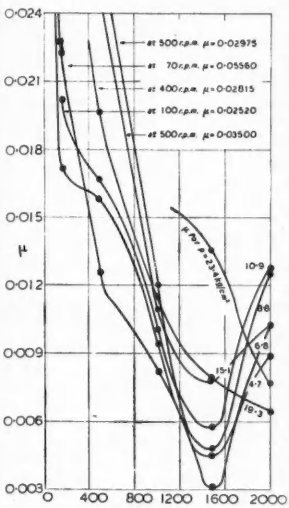


Fig. 8.

kg./cm².) and the speed low (5 m. sec.) in rolling mills, the load is only intermittently applied. Consequently a wear of the bearing does take place, but is of a lower rate than with metallic bearings.

Fig. 9 shows the limiting load and speed conditions for *L 30 Type 2* plastic bearings, similarly to Fig. 7 for Mipolam. This diagram might serve as a useful guide for design engineers. It shows clearly the maximum permissible loads up to speeds of 6 m. sec. The viscosity and rate of oil flow must, of course, be considered. Summarizing, it can be stated that with thermosetting materials a maximum load of 50 kg./cm². should not be exceeded, and in practice designs should be based rather on 30-40 kg./cm².

Improvements can be made by applying the plastic material on to the shaft, or if it would be possible to use metal bearings with thin layers of plastic coatings in order to reduce swelling.

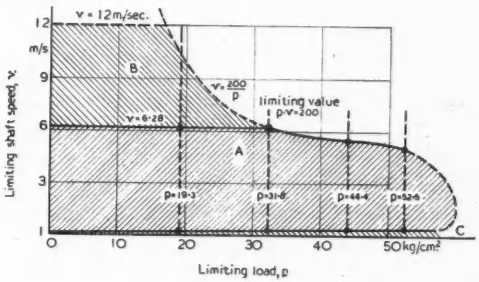


Fig. 9.

THE OERLIKON ELECTRO-DYNAMOMETER BALANCE

By W. Borel and W. Kessler. (From *Bulletin Œrlikon*, No. 246, November/December, 1943, pp. 1576-1580.)

When used as a generator, the electro-dynamometer balance serves to determine the power output and the efficiency of a prime mover, such as an internal combustion engine, an electric motor, or a turbine and the like. *Vice versa*, when the dynamometer is used as a motor, it enables the determination of the power consumption of all kinds of power consuming revolving machinery, such as pumps, compressors, machine tools, etc. Owing to its superior accuracy, and its ease of application, the electro-dynamometer balance is not only employed for purposes of scientific research, but also for rapid routine testing of machines produced in series, as illustrated in Fig. 1.

A dynamometer may be built for either alternating current or for direct current operation. In electrical respects its working principle is that of an ordinary electric generator or motor respectively, its design features differing from those of the ordinary electric machine only in so far as the stator is mounted on trunnions of the ball-bearing type. As will be seen in Fig. 2, the stator is equipped with levers carrying travelling poises similar to those of an ordinary steelyard. The position of these poises must be adjusted to bring the levers back into the zero (level) position when the dynamometer is under the test load. In the zero position the product of lever length times weight will

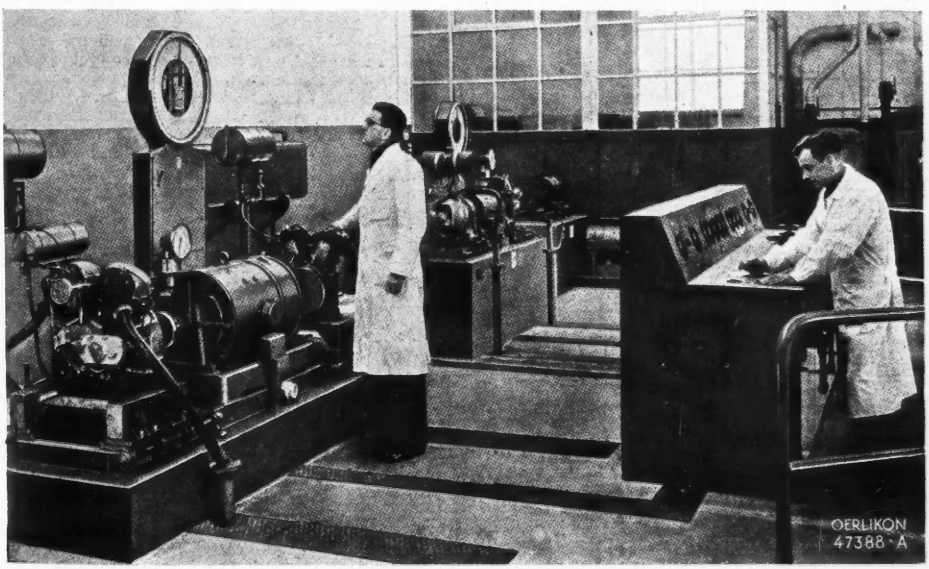


Fig. 1.

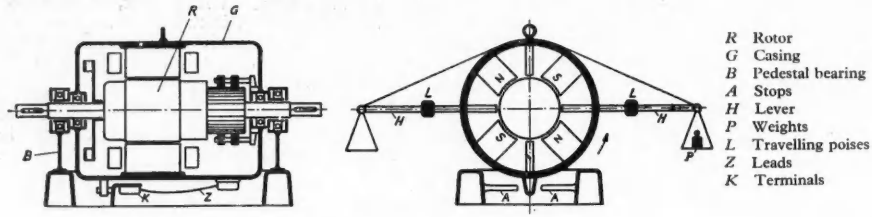


Fig. 2. **Schematic diagram of Electro-Dynamometer Balance.**

then equal the torque either exerted by the prime mover to be tested, or exerted by the dynamometer upon the power consuming machine, as the case may be.

As the power losses originating in a mechanical transmission between dynamometer and the tested machine cannot be measured separately by the dynamometer, direct coupling between machine and dynamometer should be resorted to. In cases where rapid measurement is essential, a dynamometer combined with an automatic precision balance of the compensating double-pendulum type may be used. In this type of balance, which is illustrated diagrammatically in Fig. 3, the deadweight of the lever system is compensated for by means of the adjustable counterweight *G*. The force acting upon the lever of the dynamometer is seen to be transmitted to the lever system of the balance *via* the knife-edge *B*. A reversing lever *U* is provided to effect uni-directional deflection of the index arm regardless of whether the balance is loaded by tension or by thrust. Vibrations of the index arm are deadened by means of the dashpot *D*.

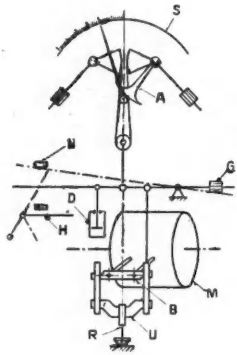


Fig. 3. **Lever system of double pendulum compensating balance.**

In order to obtain smooth and accurate indication the balance is provided with a pre-loading weight the effect of which is cancelled out by setting the balance correspondingly. If the dynamometer is to be operated in a rotational direction opposite to that for which the balance is set in this manner, the pre-loading must be correspondingly reversed. This is done by placing a deadweight upon the stator body (Fig 4) thereby pre-loading the balance to an equal extent in the opposite direction.

In operation, the stator will be rotated from its original level position to an extent determined by the magnitude of the torque prevailing. But owing to the large leverage of the balance, this rotation will not exceed an angularity of 1°30′ at maximum torque. The effective leverage will not, therefore, be reduced to less than 0.9997 times its original length at maximum torque, which is a negligible reduction. A slight inaccuracy is, however, introduced by the reaction of the cooling air discharged by the dynamometer to atmosphere. This reaction is not transmitted to the stator nor is it registered by the balance. It is kept to a minimum by appropriate arrangement of the cooling air discharge. Actually, this error amounts to about 0.2 per cent in the case of a 12 kW. dynamometer turning at 1,500 r.p.m. This error may be neglected in the case of small machines; but where highest measuring accuracy is essential—and especially where this is required at high rotational speed—a chart indicating the required correction may be used.

The total error incurred amounts to some 0.1 per cent of the design torque of the dynamometer. If torques considerably smaller than the design torque are to be measured, this error will be correspondingly increased. Where widely differing torques are to be measured with highest accuracy, the provision of a set of dynamometers with complementary operating ranges is therefore to be recommended. For a given test, dynamometer capacity should be such that its operating range need not be extended below about 30 per cent. of the torque for which the dynamometer is designed.

For a.c. operation, the dynamometer is equipped with either a squirrel cage motor or with a slip-ring motor, the latter offering the advantage of speed control. But two-, three- or four-speed dynamometers with special squirrel cage motors are also built. Generally, the employment of a.c. dynamometers will be resorted to only in cases where it is to operate as motor. In contrast to this, the d.c. type may be operated with equal facility either as motor or as generator. Where small speed ranges are concerned, a machine with self-excitation will prove satisfactory. Better stability of operation over wide ranges of load and speed will be obtained with the use of an auxiliary exciter.

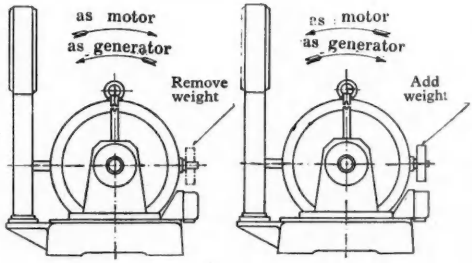


Fig. 4. **Adding or removing pre-loading weight according to the direction of rotation.**

NEW ELECTRIC DRIVE FOR ROLLING MILLS

By P. A. FRIDKIN. (From *Stal*, Moscow, Vol. 1943, Nos. 9/10, October, 1943, pp. 30-36).

A NOVEL type of electric drive has been developed which is equally well suited for new rolling mill installations and for the re-equipment of existing mills equipped with out-of-date drives. This new drive can be best described by referring to a specific case in which the mill was driven by a steam engine of 1100 i.h.p. turning at 75 r.p.m. The old drive had to be urgently replaced by a more powerful electric drive to be rated at 1500 kW. Replacement of the steam engine and its flywheel by an electric drive of the conventional type would have necessitated the shutdown of the mill for several months, since the old foundations would have had to be replaced by new ones for the motor.

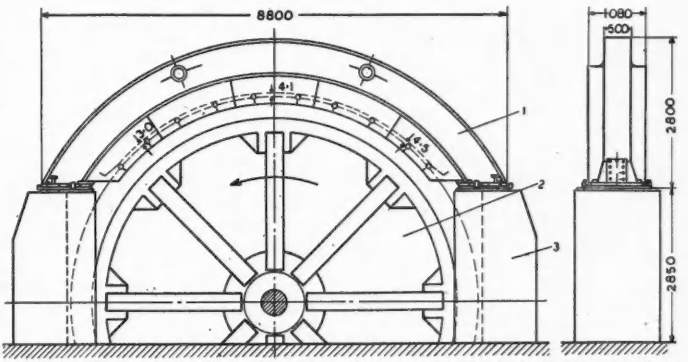


Fig. 1. **New type of rolling mill drive.** (1) Stator (2) Flywheel (3) Stator foundation

With the employment of the novel type of drive, however, the old flywheel could be made to serve as rotor of the new drive, as shown in Fig. 1. Here the stator is seen to have the shape of an arc encompassing only the top part of the periphery of the flywheel rotor. This stator is equipped with 11 pairs of poles, the radius of the pole faces being 3822 mm. and the arc of the stator including an angle of 105 degrees. In electrical respects this machine is equivalent to an ordinary stator with a radius of $r = 3822 \cdot \frac{105}{360} = 1115$ mm. with a speed of $n = \frac{60.f}{p} = \frac{60.50}{11} = 273$ r.p.m.

A considerable magnetic pull amounting to 40 tons is exerted upon the magnetised flywheel and thus supports in operation some 67 per cent of the total weight of the revolving mass. The synchronous speed of the machine is given by the equation $n = \frac{60 \cdot \tau \cdot f}{\pi R}$ where τ is the pole pitch, f is the network frequency and R is the radius of the pole faces. As in the case considered it is $\tau = 300$ mm. and $R = 3822$ mm., the synchronous speed is $n = \frac{60.300.50}{\pi \cdot 3822} = 75$ r.p.m. The turning force per sq. cm. of active stator surface is given by the equation

$$f_t = 0.51\, B_e^2\, v_s \frac{r_{2s}}{(z_{2s})^2} \cdot 10^{-15}\ kg./cm.^2$$

where B_e is the amplitude of the sinusoidal induction curve in the air gap, v_s is the linear speed of the slip in cm. per second and r_{2s} and z_{2s} are respectively the resistance corresponding to the slip and the entire resistance of the rotor surface (Ohm per sq. cm.). In the case considered with a load of 1500 kW. it is $B_e = 6900$, $v_s = 63$ m. per sec., $r_{2s} = 8.05.10^{-6}$ Ohm per cm.^{2} and $z_{2s} = 8.1 \cdot 10^{-6}$ Ohm per sq. cm.

Therefore

$$f_t = 0.51 \cdot 6900^2 \cdot 63 \frac{8.05 \cdot 10^{-6}}{(8.1 \cdot 10^{-6})^2} \cdot 10^{-15} = 0.188 \text{ kg. per sq. cm}$$

The difference between the values of r_{2s} and z_{2s} will increase with diminishing slip v_s of the rotor. The total force developed by the stator is given by

$$F_t = 0.51\, B_e^2\, v_s \frac{r_{2s}}{(z_{2s})^2} \cdot S \cdot 10^{-15}\ kg.$$

where S is the active surface of the magnetic part of the stator in sq. cm. The power of the developed machine is then found as $P = 0.5\, B_e^2\, v_s \frac{r_{2s}}{(z_{2s})^2} \cdot S\, v_r \cdot 10^{-17}$ kW.

where v_r is the actual linear speed of the rotor surface taken at the radius of the pole faces of the stator in m. per second. The magnetic pull exerted during operation by the stator upon the rotating part of the machine is given

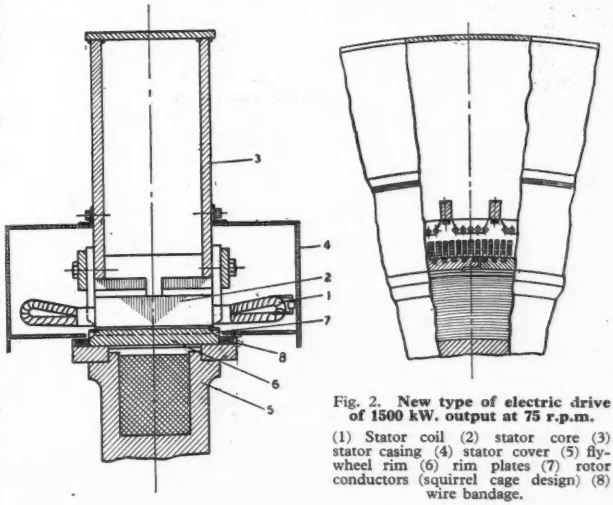


Fig. 2. **New type of electric drive of 1500 kW. output at 75 r.p.m.**

(1) Stator coil (2) stator core (3) stator casing (4) stator cover (5) flywheel rim (6) rim plates (7) rotor conductors (squirrel cage design) (8) wire bandage.

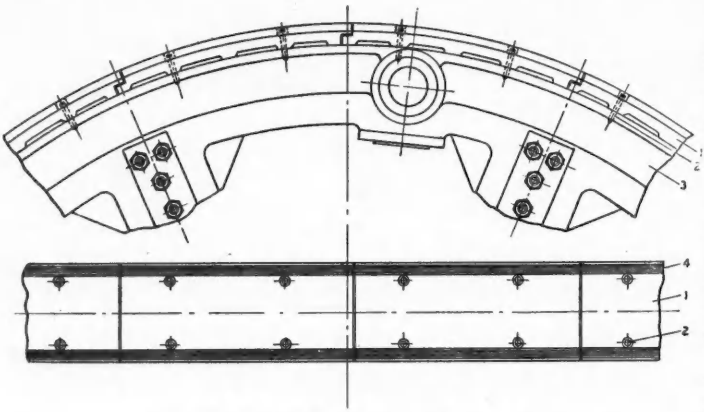


Fig. 3. **Detail of flywheel conversion to rotor.**
(1) Rim plates (2) steel bolt (3) rim of flywheel (4) wire bandage.

by $F_m = \frac{1}{2}\left(\frac{B_e}{5000}\right)^2 . k . S . \frac{2 \sin \varphi/2}{\varphi}$ kg., where k is the Carter coefficient and φ is the angle included by the arc of the stator.

The stator of the machine consists of the four principal elements shown in Figs. 1 and 2, namely: (1) insulated stator coils; (2) laminated stator core of 0.5 mm. electric sheet; (3) the stator casing of welded construction bolted to the concrete foundation; (4) segments of sheet iron of 1.5–2.0 mm. thickness covering the core faces. The coil ends are securely held in place to avoid deformation by electro dynamic stress. Provision is made to vary the air gap by plus or minus 50 per cent of its normal value. By the employment of the novel system of axial ventilation, first introduced in 1940, uniform and intensive cooling is achieved.

The total weight of the stator amounts to 10.8 tons, of which 0.8 tons comprise the weight of the copper and 3.0 tons that of the core. The weight of the casing is 5.8 tons,welded plate segments weighing another 0.4 tons and other components amounting to 0.8 ton in weight.

The rim of the flywheel shown in Fig. 2 is equipped with a wire bandage, a new steel rim having been added on top of the old one. Because of the considerable circumferential length of this rim (24 m.), the new rim was made of a number of separate pieces fastened to the old rim by means of two wire bandages, as shown in Figs. 2 and 3. The bolts (2) serve to keep the rim plates in place while applying the bandages. Moreover, they keep the rim plates from shifting upon the flywheel rim under the forces exerted by the stator or by the forces of acceleration. The compression of 140 tons exerted by the bandages upon the rim is equivalent to a frictional force of 28 tons; and this alone is large enough to keep the rim plates from moving.

The construction of the joints between the rim plates (Figs. 2 and 3) allows for thermal expansion of the plates, at the same time maintaining an unbroken magnetic flux between adjoining plates. The temperature rise of the plates is computed as 30 deg. C. above ambient air temperature.

The rim plates of the rotor contain axial slots which are approximately 1.4 × 2.0 cm. in section, into which steel and copper conductors are inserted. These are short circuited by copper end connections of about 2 sq. cm. in cross section. The purpose of the copper rods is to improve the electric contact between the steel bars and the rotor plates. The total weight of the rim is 8.8 tons including the weight of the wire bandage, which amounts to 0.45 tons and that of copper which totals some 0.3 tons. The chief data of this novel machine are given in the subjoined table:

TABLE I

	Conventional electric motor for direct drive	Conventional electric motor with reduction gear drive	New type of electric drive
Rated output, kW.	1500	1500	1500
Speed, r.p.m.	75	490/75	75
Voltage	6,000	6,000	6,000
Frequency, cycles	50	50	50
Number of pole pairs of stator . .	40	6	11
Number of coils in stator . .	720	84	204
Rated efficiency, per cent . .	87	94·97	93
Power factor	0.55	0.88	0.70
Weight of drive, tons	52	84	22
Copper requirements, tons . .	2.1	1.45	1.1
Electric sheet requirements, tons . .	6.7	3.8	3.0
Decrease in frictional losses in bearings owing to magnetic upward pull, kW.	—	—	12.0
Lubricating oil consumption, tons per year	1.0	2.5	0
Number of couplings required . .	1	2	0
Relative time required for manufacture and erection of drive . .	100	145	42
Shutting down period of mill for erection of drive, working days	55	70	4

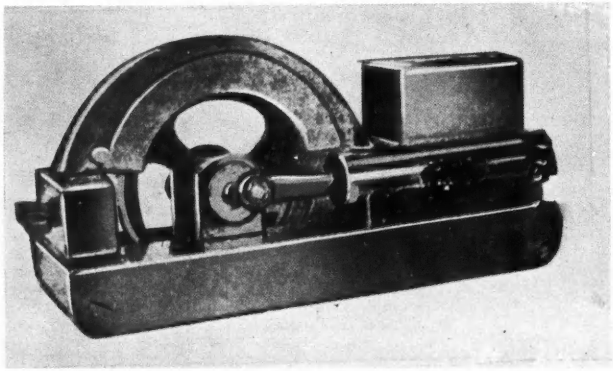


Fig. 4. **Reciprocating pump equipped with novel drive.**

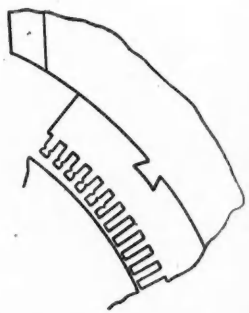


Fig. 5. **Configuration of end of pole face.**

The magnetic upward-pull actions on the rotor amount to 67 per cent of the weight of the latter, and decreases the load acting on the shaft and also on the bearings. The nominal turning force reacting upon the stator amounts to 5.2–6 tons uniformly distributed over its two arches, each of which is some 7.0 m. in length, thus giving a force of about 7.5 kg. per cm. of length of the arch. At the peak loading of 3500 kW. the linear loading rises to 18 kg. per cm. as compared to a corresponding force of 30 tons concentrated upon the pinion of the reduction gear of the orthodox motor drive, if such were employed. The absence of the conventional reduction gears, and so of all the attendant mechanical problems and troubles, is an outstanding feature of this novel drive. The higher the linear speed of the rotor, the greater will be the freedom of layout afforded to the designer. This is exemplified by the data listed below, which show the improvement in electrical and mechanical characteristics that can be obtained with higher linear speeds. The smallest machine listed was built in 1939 and serves as drive for a pulveriser mill of the ball type. Another machine for the same purpose which is under construction will be equipped with 30 pole pairs, its specified speed being 24 r.p.m. Its electrical and mechanical characteristics will be superior to that of the 1939 model.

The 280 kW. unit listed, serves as a drive for a reciprocating pump, as shown in Fig. 4. The 1500 kW. unit shown in Table II, represents the roll mill drive referred to above. The high peripheral speed employed for the flywheels of rolling mills, compressors, etc., enables the choice of motor dimensions conducive to low weight and to optimum electric characteristics.

TABLE II.

	Ball mill drive	Reciproc. pump drive	Roll mill drive
Rated output, kW.	240	280	1,500
Voltage	3,000	2,000	6,000
Normal speed, r.p.m.	29.8	59.3	75
Frequency, cycles	50	50	50
Normal peripheral motor speed, m./sec.	6.15	7.65	30
Radius of rotor, mm. ..	1965	1238	3817
Air gap, mm.	2.8	1.6	4.5
Unit weight of stator, kg./kW.	2.9	1.3	0.75
Rated efficiency, per cent ..	76	84	93
Power factor	0.38	0.53	0.7
Number of pole pairs	48	20	11

The configuration of the pole face at the ends of the arch of the stator core is outlined in Fig. 5. This shape ensures a gradual decrease in the magnetic flux towards the ends of the core.

The advantages of this new type of drive as applied to the modernisation of rolling mills can be summarised as follows: The change-over from steam operation to electric drive can be much more quickly effected than with the conventional drive consisting of motor and reduction gear. In the instance referred to, replacement was effected in 106 days. In such a relatively short period of time, removal of the old steam engine and its replacement by a conventional type of electric drive could not possibly be carried out.

ESTABLISHING LISTS OF TYPES IN THE MANUFACTURE OF THREE-PHASE MOTORS

By F. GÖTZ and B. CERNAVIN. (From *Elektrotechnische Zeitschrift*, Vol. 65, Nos. 21/22, June 1st, 1944, pp. 207-212).

IN establishing lists of types of electric motors, the two prime requisites to be met are (1) minimum cost of materials and (2) low manufacturing cost. That these two requirements oppose each other will be seen from the following example: It may be assumed that a list of types of X motors of different capacities is to be computed, all types to have identical numbers of poles and to operate at identical voltages. For each one of these motors a design can be found which requires a minimum amount of materials. But this would mean that every one of these motors would be of a different design. Thus there would be required X complete sets of castings, X sets of dies, and X complete sets of jigs, and, while requiring a minimum amount of materials, this list of types would nevertheless show highest manufacturing cost.

In order to simplify the production of lists of types, machines graduated in size should have as many identical component parts as possible; and in order to diminish the number of non-active parts, groups of machines must be formed in which the variations in output are obtained by increasing the length of the core, with the stator diameter remaining unchanged. One single punching would be used for the machines of such a group. In this way a minimum number of punchings for the entire list of X machines would be obtained.

It is quite obvious that this kind of solution will enforce deviations from optimum dimensions as far as electrical aspects are concerned, but it leads to simplifications of production. In this connection it must not, however, be overlooked that the larger the quantity to be produced of a list of types, the smaller are the deviations from optimum dimensions that can be permitted; or, in other words, the larger must be the number of stator diameters and the smaller must be the variations in the length of the stator.

The growing employment of single drives has lead to the manufacture of much larger series of motors of capacities up to 10 kW than of larger machines. From the economic point of view it is therefore preferable to design the lists of types of machines up to 10 kW capacity for a large number of stator diameters, keeping the different widths of the stators correspondingly small, and usually co-ordinating two to three different stator widths with one single diameter. Where lists of types exceeding 10 kW in capacity are concerned, it is preferable to co-ordinate three to five stator widths with one stator core diameter.

Consideration must also be given to the fact that a list of types may contain machines of different speeds. But it is a well known fact that for each engine speed perfect utilisation of the material is conditional upon adherence to a co-ordinated ratio of external and internal diameter of the stator core. This can be proven by considering that the magnetic flux

(1) $\Phi = B \tau L$ where B is the average induction in the air gap, τ is the pole pitch, and L is the width of the stator core. But it is also

(2) $B_j = \Phi / 2 L h_j k_f$ where B_j is the induction in the yoke of the stator core, h_j is the height of the yoke and k_f is a factor which takes account of the insulation of the electrical sheet. Combining equations (1) and (2) it can be written:

(3) $h_j = B\tau / 2 B_j k_f$.

This shows that by increasing the number of poles and consequently reducing the pole pitch, the height of the yoke is decreased. But it means also that the ratio of outer stator diameter to inner stator diameter is decreased with increasing number of poles.

To each number of poles corresponds a certain ratio of outer and inner diameter of the stator core, so that there exists a co-ordinated inner diameter for each number of poles, if the outer stator diameter is kept unchanged. But if for each engine speed the optimum inner diameter is chosen, a large number of different punchings will be needed, thereby complicating production and increasing production cost. It is for this

reason that where large numbers of poles are concerned, and where the difference between optimum internal diameters are comparatively small, identical inner stator core diameters are chosen.

Assuming for instance an outer core diameter D_0, the internal diameter may be fixed as follows:

D_1 is the smallest inner diameter and is to be used for a number of poles $2p = 2$.

D_2 is larger than D_1 and is to be used for $2p = 4$,

D_3 is larger than D_2 and is to be used for $2p = 6$,

D_4 is larger than D_3 and is to be used for $2p = 8, 10, 12$.

Assuming further that three different stator core widths are to be used with any one identical outer diameter D_0 and with four different internal diameters, as listed above, 18 different machines with identical non-active component parts are obtained, viz.:—

3	different capacities	with	$2p = 2$,
3	,,	,,	,, $2p = 4$,
3	,,	,,	,, $2p = 6$,
9	,,	,,	,, $2p = 8, 10, 12$.

This constitutes the minimum "cell" of a list of types for a given voltage, comprising, as it does, 18 different motor types with different capacities and speeds. If different modes of cooling and protection are to be included in the production programme, and if widely different voltages are to be provided for, etc., the number of types will be correspondingly larger.

With regard to factors which limit the production of motors in lists of types, it has been found in practice that the larger the diameter of the machines the larger will be the various numbers of poles for which the motors can be built. The maximum number of poles for which a machine of a given capacity can be built is determined by the minimum pole pitch which is practically feasible. To retain satisfactory electric characteristics, a minimum of two slots per pole and phase is required. To give satisfactory results, the width of slot and tooth must not be less than 1.1 cm.; therefore the minimum pole pitch feasible amounts to $\tau = 13.2$ cm., but a $\tau = 7$ cm. is still permissible for half-open slots.

In the case of machines of the open type an upper limit exists from the aspect of ventilation noise. According to Trapeznikov, this upper limit occurs at 45 m./sec. For larger peripheral speeds, totally enclosed machines with forced ventilation must be used.

When determining the various widths of stator core to be used with any one external core diameter, care should be taken to make the smallest width to be used with one diameter equal to the largest width of the next-smaller diameter, and so forth. In analogous manner the number of inner and outer diameters can be reduced to a minimum.

A special method for the computation of three-phase induction motors developed by Trapeznikov rests on the use of characteristic motor parameters, such as pole pitch and circumferential speed. It can be easily proven that the losses, the amount of active materials required, the magnetizing current, and certain other characteristic dimensions of a motor are exclusive functions of pole pitch or of peripheral speed.

Considering, for instance, two machines of different diameters but of identical pole pitch, as indicated in Fig. 1, the characteristic properties of these two machines being assumed as identical: The stator core of the larger right-hand machine can easily be visualised as being built up of eight times the punchings of the smaller machine, so that the only difference in the behaviour of the two machines will be the engine speed. This shows that the pole pitch is an important machine parameter, which is to say that the characteristics of asynchronous induction motors are defined by those of an elementary machine of corresponding pole pitch. Differences in the electrical characteristics of machines of identical pole pitch but of different speeds, will be due to secondary factors such as differences in the contour of the teeth due to the differences in the stator core diameters and differences in the air gaps due to considerations of manufacture.

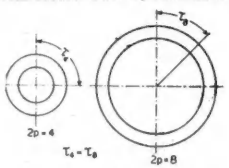


Fig. 1. **4- and 8-pole motors with identical pole pitch.**

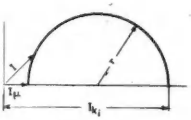


Fig. 2. **Circle diagram of three phase induction motor.**

In polyphase induction motors, leakage is known to be a factor of great importance, since it serves to determine the power factor cos φ and with it the overload capacity of the machine. From the elementary circle diagram given in Fig. 2, it is seen that the largest possible power factor and also the power factor at nominal rating is a function of the ratio

$$\sigma = \frac{I\mu}{I_{k1}} \quad .. \quad (4)$$

where σ is the leakage factor, $I\mu$ is the magnetizing current, and I_{k1} is the idealised short circuit current. It can also be written

$$\sigma = \frac{I\mu}{I_{k1}} = \frac{X_k}{X_m} \quad .. \quad (5)$$

where X_m is the reactance of the main flux and X_k is that of the leakage flux.

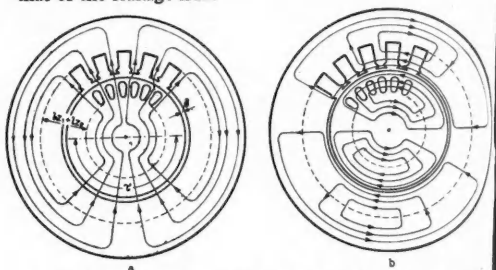


Fig. 3. (a) **Lines of magnetic main flux** (b) **Lines of magnetic leakage flux.**

The lines of the main magnetic flux are indicated in Fig. 3a; while those of the leakage flux are shown in Fig. 3b. The reactance of a single phase of the main flux can also be expressed by

$$X_m = \frac{w^2 \cdot \tau \cdot f \cdot L}{\delta' \mu p} \quad .. \quad .. \quad .. \quad (6)$$

where w denotes the number of turns per phase, δ' is the idealised air gap and

$$\mu = \frac{\text{number of magnetic circuit ampere-turns}}{\text{number of gap ampere-turns.}}$$

The leakage flux can be expressed by

$$\Phi_\sigma = \frac{(l_{z_1} + l_{z_2})\, A\, W_s}{\tau \cdot k_1} L \quad .. \quad .. \quad (7)$$

where l_{z_1} is the height of the teeth of the stator, l_{z_2} is that of the teeth of the rotor, AW_s denotes the ampere-turns of the leakage flux, and k_1 is the average ratio of slot width/slot pitch for both stator and rotor. The reactance of the slot leakage flux is

$$X_\sigma = \frac{w^2 \cdot f\,(l_{z_1} + l_{z_2})}{\tau \cdot k_1} \cdot \frac{L}{p} \quad .. \quad .. \quad (8)$$

The coil-end leakage flux and the flux of the harmonics

contribute relatively little to the total leakage, so that $X_\sigma = X_k$, and it is approximately

$$\sigma = \frac{I_u}{I_{k1}} = \frac{X_k}{X_m} = \frac{w^2 . f (l_{z_1} + l_{z_2}) \delta' . \mu}{\tau . k_1 . w^2 . \tau . f} = \frac{(l_{z_1} + l_{z_2}) \delta' . \mu}{k_1 \tau^2} \quad (9)$$

POLE PITCH vs. MAGNETISING CURRENT.

The magnetising current is given by the expression

$$I_u = \frac{1.11 . AW . 2 . p}{m . w . fw} \quad .. \quad .. \quad (10)$$

and it is also

$$AW = \frac{0.8 . \delta' . B_{m1} . \mu . L}{\alpha . L_1} \quad .. \quad .. \quad (11)$$

and $$w = \frac{\pi . D . AS}{2 . m . I} = \frac{\tau . p . AS}{m . I} \quad .. \quad .. \quad (12)$$

where AW denotes the ampere-turns of the magnetic circuit, δ' is the idealised air gap, τ is the pole pitch expressed in cm., p is the number of pole pairs, D denotes the inner diameter of the stator expressed in cm., L is the net length of the laminated core excluding the ventilating ducts, L_1 is the gross width of the laminated core including the ventilating ducts, AS is the linear loading in amps. per cm., fw is the winding factor, and

$$\alpha = \frac{b_1}{\tau} = 0.69.$$

The mean value for the induction in air during idling of the machine can be expressed by

$$B_{m1} = \frac{\Phi}{\tau . L} \quad .. \quad .. \quad .. \quad .. \quad (13)$$

and it therefore becomes

$$\frac{I_u}{I} = \frac{2.55 . \delta' . \mu B_{m1} . L}{f_w . \tau . AS . L_1} \quad .. \quad (14)$$

Taking the usual values for f_w, μ, AS, and B_{m1}, the relationship $\frac{I_u}{I} = f(\tau)$ for different values of δ' can be charted as shown in Fig. 4; while the relationship of I_u and σ *versus* the pole pitch can be used to chart the functionality $\cos\varphi = f(\tau, \delta)$ as shown in Figs. 5 and 6. This shows that the power factor of the induction motor does not depend upon the output but is a function of the pole pitch and of the air gap.

According to Trapeznikov, the mechanical losses relative to the peripheral speed are as follows :

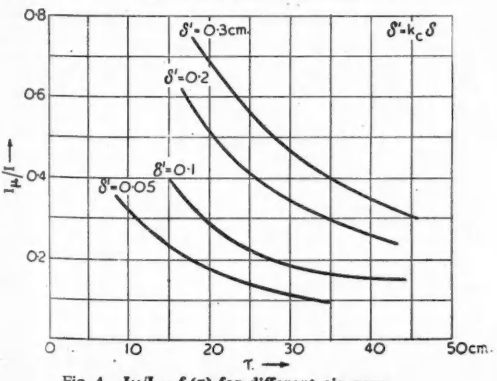


Fig. 4. $I\mu/I = f(\tau)$ for different air gaps.

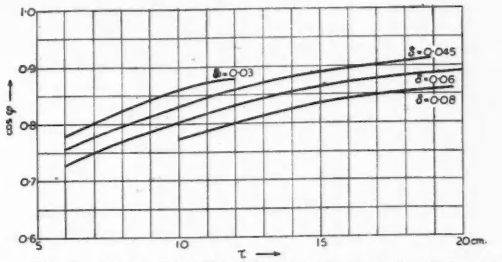


Fig. 5. $\cos\varphi = f(\tau)$ for different air gaps for induction motors with half-closed slots.

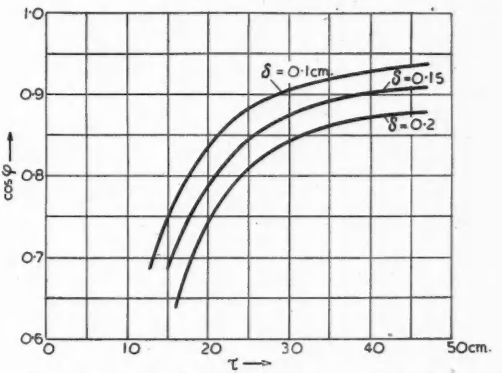


Fig. 6. $\cos\varphi = f(\tau)$ for different air gaps for induction motors with open slots.

v [m./sec.]	W_{mech} [%] cos φ
15	0.65—0.45
25	0.9 —0.6
35	1.1 —0.75
45	1.3 —0.9

where the higher values refer to motors with great stator length. In the case of 4-pole machines the losses range 20-30 per cent higher than listed above.

Core Losses.—The losses in the teeth based on the number of teeth per pole amount to

$$\frac{Q_z}{2p} = k_z V_z . B_z^2 \ (15) = k_z \frac{C_z}{\alpha} \tau^2 . L . B_{m1} . B_z \quad (16)$$

where B_z is the flux density in the teeth of the stator $C_z = l_{z_1}/\tau$, and k_z is a coefficient which takes account of the quality of the electric sheet and also of the frequency. The losses in the yoke are given by

$$\frac{Q_j}{2p} = k_j \frac{C_j}{2} \tau^2 . L . B_{m1} B_j \quad .. \quad (17)$$

where $C_j = D_j/D$ (see Fig. 7).

The total core losses, therefore, are

$$\frac{Q_{Fe}}{2p} = \tau^2 L . B_{m1} \left[k_z \frac{C_z}{\alpha} B_z + k_j \frac{C_j}{2} . B_j \right] \quad .. \quad (18)$$

For a given quality of electric sheet and for a given frequency this expression can be simplified thus

$$\frac{Q_{Fe}}{2p} = A . \tau^2 . L . B_{m1} \quad .. \quad .. \quad (19)$$

where $$A = k_z \frac{C_z}{\alpha} B_z + k_j \frac{C_j}{2} . B_j \quad .. \quad (20)$$

It is preferable to express the total core losses in per cent of the power input of the machine by writing

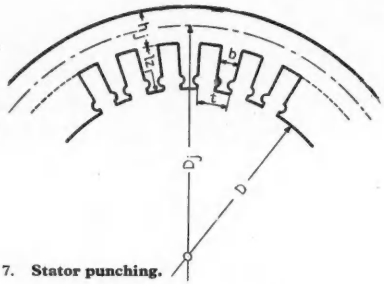


Fig. 7. **Stator punching.**

$$\frac{P}{2_p} = \frac{5 \,.\, 10^{-8} \,.\, k \,.\, AS \,.\, B_{m1} \,.\, L \,.\, u^2}{f} \quad .. \quad (21)$$

where P is the power input in kVA, u is the peripheral speed in m./sec., $k = f_s \,.\, f_{w_2} \, f_s$ is the magnetic field form factor or belt differential factor, and f_w is the winding factor. The percentage core loss thus becomes

$$\overset{[\%]}{Q}_{Fe} = \frac{A \,.\, \tau^2 \,.\, L \,.\, B_{m1} f \,.\, 50^2 \,.\, 100}{5 \,.\, 10^{-8} \, k \,.\, B_{m1} \, AS \,.\, L \,.\, \tau^2 f^2 \cos \varphi \,.\, 10^3} \quad (22)$$

or

$$\overset{[\%]}{Q}_{Fe} = \frac{C_{Fe}}{AS \cos \varphi} \quad .. \quad .. \quad .. \quad (23)$$

where the constant C_{Fe} characterises the conditions of the magnetic circuit, the quality of the electric sheet and also the accuracy of its assembly. Computed values of C_{Fe} can be easily verified by actual measurement of Q_{Fe}. The discrepancy between computed and measured values of C_{Fe} will be of the order $\pm$ 15 per cent. Average values for C_{Fe} obtained by test are given in the subjoined table, which is based on a value of $f = 50$ Hz, an electric sheet loss of 3.3 W per kg. and an induction $B = 10{,}000$ Gauss.

Average Values for C_{Fe}.

Type of Slots		
Stator	Rotor	C_{Fe}
Open	Half closed	800—950
Half closed	„ „	650—700
„ „	Closed with Al cage	500—575

Copper Losses.—The copper losses in the stator are given by :

$$\overset{[\%]}{Q}_1 = \frac{s \,.\, \rho \,.\, 10^6}{u \, B_{m1} \cos \varphi} \left(\lambda + \frac{\beta_s}{\gamma} \right) \quad .. \quad (24)$$

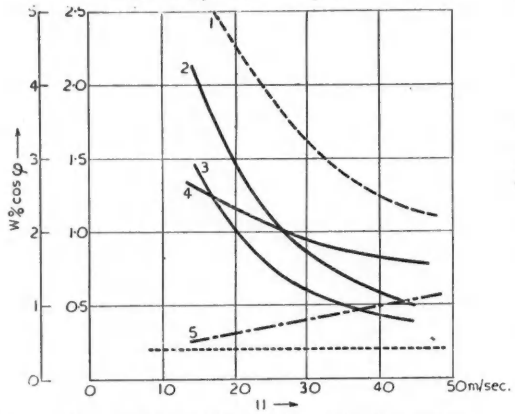


Fig. 8. **Segregated losses of three-phase induction motors of the slip ring type.**

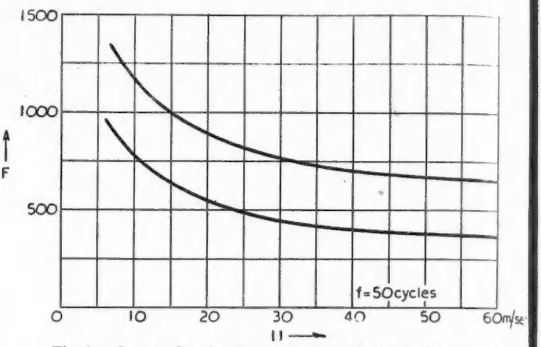


Fig. 9. **Curves for the determination of factor F in equation (27) [Determination of total core weight].**

where s is the current density in the conductors in amp. per sq. mm., ρ is the specific resistance of the conductors in ohm per sq. mm., $\lambda = \dfrac{l_1}{L}$ is the ratio of *core length with ventilation ducts/core length without ventilation ducts*, $\beta_s = \dfrac{l_s}{\tau}$ is the ratio of *length of end windings/pole pitch*, and $\gamma = \dfrac{L}{\tau}$ is the ratio of *core length without ventilating ducts/pole pitch*. The copper losses of a squirrel cage rotor are found from

$$\overset{[\%]}{Q}_2 = \frac{s \,.\, \rho \,.\, 10^6}{u \,.\, B_{m1} \cos \varphi_{\text{eff.}}} \left(\lambda + \frac{2 \, l_s'}{L} + 2/\pi \gamma \right) \quad (25)$$

where l_s' is the length of the projecting part of the conductors.

PERIPHERAL SPEED vs. EFFICIENCY.

The influence of the peripheral speed upon the various losses of a slip ring motor is exemplified in Fig. 8, where the losses are shown in per cent of the motor output multiplied by cos φ in order to eliminate the influence of the power factor. Curve 1 in Fig. 8 represents the ratio of *copper losses/core losses*, curve 2

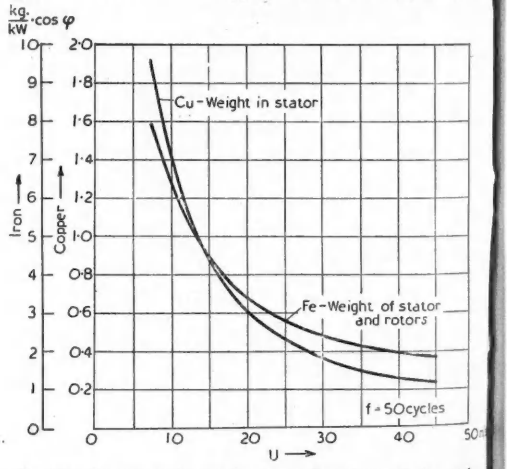


Fig. 10. **Copper and core losses or functions of the peripheral speed of a list of types.**

represents the copper losses in the stator, and curve 3 those in the rotor. The total core losses are given by curve 4, the mechanical losses by curve 5, and the sundry losses by curve 6. These curves apply to three-phase motors of the slip ring type characterised by $f = 50$, $2p = 6$, $s_1 = 4.75$ amp. per sq. mm., and $s_2 = 4.5$ amp. per sq. mm. From this chart the conclusion can be drawn that in the overloading of machines of *high* peripheral speed, the efficiency of the machine either remains unchanged or it increases. But in the case of machines with *low* peripheral speed, overloading will lead to a rapid deterioration in efficiency. The amount of stator copper required in relation to the power input is given by

$$\frac{G_{cu}}{P} = \frac{8 \,.\, 9 \,.\, 10^4}{u \,.\, B_{m1} \,.\, s \,.\, \cos\varphi} \left(\lambda + \frac{\beta_s}{\gamma}\right) [kg./kW] \qquad (26)$$

while the combined electric sheet requirements of stator and rotor (incl. the wastage incurred in the punching of the slots) can be taken to approximate to

$$\frac{G_{Fe}}{P} = \frac{F}{AS \cos\varphi} \quad .. \quad .. \qquad (27)$$

values for F being given in Fig. 9. The core and copper losses computed for a list of types of machines are charted in Fig. 10.

DIAGRAMS FOR QUICK DETERMINATION OF ANGLES REQUIRED FOR LAYING OUT SHEET METAL WORK.

By C. J. WALBAUM, VDI. (From *Zeitschrift des VDI*, Vol. 87, No. 33/34, August 21st, 1943, p. 540).

IN laying out sheet metal work such as containers or bunkers in the shape of truncated pyramids, the various angles of inclination or intersection of the component sheets must be determined. While hitherto it has been usual practice to determine these angles by computation, or on the drawing board, the simple method given below permits their determination by means of charts.

Referring to Fig. 1, the angle γ can be expressed by the equation $\cot\gamma = \sqrt{\cot^2\alpha + \cot^2\beta}$. On the basis of this relationship γ can be determined from the chart Fig. 2 in the following manner: By way of example and assuming $\alpha = 50$ deg. and $\beta = 30$ deg. A is found as the point of intersection of the y-axis with the slanting line representing 50 deg. Proceeding from A horizontally, point B is found as the point of intersection of horizontal and 30 deg. line. Point C is then found by projecting point B upon the abscissa X. By describing a circle with O as center, and the distance O-C as radius, point D is obtained, from whence a vertical is drawn to point F representing the intersection of the vertical with the horizontal line drawn through A. By interpolation, point E is then found to correspond to an angle of $\gamma = 27$ deg. 30 min., as indicated on the angular scale forming the top of the chart (Point F).

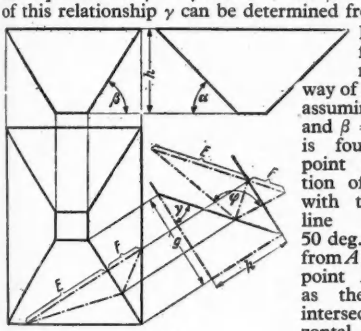


Fig. 1.

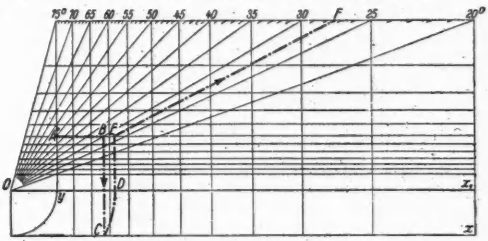


Fig. 2. **Diagram for determining γ.**

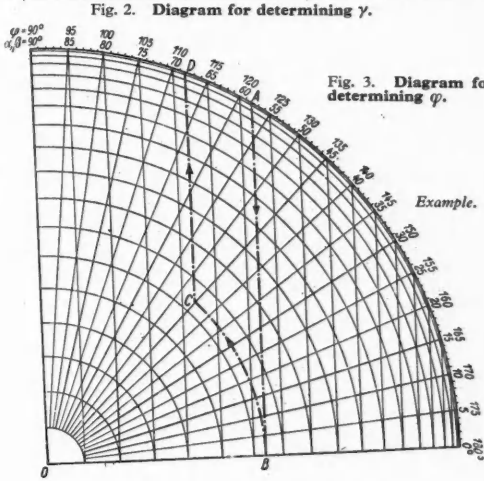


Fig. 3. **Diagram for determining φ.**

The angle φ shown in Fig. 1 can be obtained in similar manner by utilising the relation

$$\cos\varphi = -\cot\alpha \,.\, \cot\beta \,/\, \sqrt{(\cot^2\alpha + 1)(\cot^2\beta + 1)}$$

If, for instance, $\alpha = 58$ deg. and $\beta = 45$ deg., point A on the inner peripheral scale of chart (Fig. 3) will correspond to $\alpha = 58$ deg. Point B is then found as the projection of A upon the abscissa. By describing a circle with point O as centre and radius O–B, point C is found at the point of intersection of the described circle and of the radial line representing $\beta = 45$ deg. A vertical line from point C will then intersect the periphery of the graph at point D. According to the outer peripheral scale, this point corresponds to a value of $\varphi = 112$ deg.

WELDING MACHINES WITH CONDENSER DISCHARGE.

By D. GENKIN. (From *Bulletin de la Société Française des Electriciens*, Vol. 4, No. 38, June, 1944, pp. 151-160).

THE typical features of the spot welding process are the generation of heat by electric current at the point of contact resistance and the diffusion of metallic particles across the contact surface, the latter being subjected to a large force of compression. By increasing this pressure and raising the temperature, extremely short welding times can be obtained; as a matter of fact, the welding times achieved in modern practice are of the order of a few hundredths of a second. The quality of a welding process carried out in this manner will rest upon a number of factors such as maximum amplitude of the welding current, its duration, the characteristic of the current, the contact pressure, and the condition of the contact surface.

On the electrical side, the rapidity with which a spot weld is made and the relatively large interval of time prevailing between the making of the individual welds prevent the establishment of permanent periodicity. The making of a weld must therefore be considered as an event consisting of a single discharge during which both the resistance at the welding point and the induction in the core of the transformer of the welding machine are subjected to continual variations.

A welding machine operating on the principle of condenser discharge consists essentially of a transformer, the primary winding of which is passed by the discharge current of the condenser, the external secondary circuit being closed by a small impedance consisting of the ohmic resistance of the weld and the inductance of the leads carrying the welding current. The capacity of the condenser batteries employed for this purpose may amount to some tenths of a farad. In this type of welding process the transformer serves the purpose of establishing optimum conditions for the operation of the condenser battery. Obviously, without the employment of a transformer, a much larger condenser battery would be required to produce the same discharge.

Assuming that the coefficient of mutual induction and the ohmic resistance of the circuit remain constant during operation, it can be written :

$$\left.\begin{aligned} V_1 &= R_1 i_1 + L_1 \frac{di_1}{dt} + M\frac{di_2}{dt}, \\ 0 &= R_2 i_2 + L_2 \frac{di_2}{dt} + M\frac{di_1}{dt}, \\ i_1 &= -C\frac{dv_1}{dt}, \end{aligned}\right\} \quad (1)$$

where R_1 and R_2 are the ohmic resistances of primary and secondary circuit respectively, and L_1 and L_2 are the coefficients of total self-inductance including the exterior inductances. An approximate solution of these differential equations for i_1, i_2 and V_1 is

$$\left.\begin{aligned} i_1 &= -\frac{V_0 R_2{}^3 M^2 C^2}{L_2{}^4} e^{-\alpha t} + \frac{L_2 V_0}{(L_1 L_2 - M^2)\omega} \sin(\omega t + \varphi) e^{-\delta t} \\ i_1 &= \frac{V_0 R_2 MC}{L_2{}^2} e^{-\alpha t} - \frac{M V_0}{(L_1 L_2 - M^2)\omega} e^{-\delta t} \sin(\omega t + \Psi) \\ V_1 &= -\frac{V_0 R_2{}^2 M^2 C}{L_2{}^3} e^{-\alpha t} + V_0\left(1 + \frac{R_2{}^2 M^2 C}{L_2{}^3}\right)\sqrt{\frac{\delta^2+\omega^2}{\omega}} e^{-\delta t} \sin(\omega t + \lambda) \end{aligned}\right\} \quad (2)$$

where $\alpha = -\dfrac{R_2 L_2{}^2}{L_2{}^3 - R_2{}^2 M^2 C}$

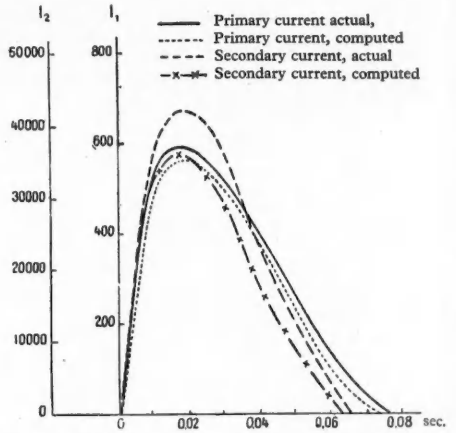
$$\omega = \sqrt{\frac{L_2{}^3 - R_2{}^2 M^2 C}{L_2{}^2 C (L_1 L_2 - M^2)} - \delta^2} \approx \sqrt{\frac{L_2}{C(L_1 L_2 - M^2)}} \quad (3)$$

$$2\delta = \frac{L_2{}^2(R_1 L_2{}^2 + R_2 M^2) - R_2{}^2 M^2 C(R_1 L_2 + R_2 L_1)}{(L_1 L_2 - M^2)(L_2{}^3 - R_2{}^2 M^2 C} \approx \frac{R_1 L_2{}^2 + R_2 M^2}{L_2 (L_1 L_2 - M^2)}$$

$$\tan\varphi \approx \sin\varphi \approx \frac{R_2{}^3 M^2 C^2 \omega (L_1 L_2 - M^2)}{L_2{}^5}$$

$$\tan\varphi \approx \sin\delta \approx \frac{R_2 C\omega (L_1 L_2 - M^2)}{L_2{}^2} \quad (5)$$

$$\tan X \approx \frac{\omega}{\delta^2}$$



These expressions show that in order to obtain the maximum effect it is necessary to make $(L_1 L_2 - M^2)$ small and to make C large. It should, however, be noted that these approximate expressions rest sufficiently accurate only in cases where the primary resistance and $R_2{}^2 M^2 C$ are small.

These approximate expressions were verified by solving a numerical problem, both by the approximate and the exact method. In order to determine the accuracy of the approximate method, the calculation was carried out with all factors being kept constant and only the coefficient M of the transformer being varied over a range from 10^{-1} to 10^{-6}.

A typical application of this method to a welding machine is as follows :

Number of primary windings ..	90
Number of secondary windings	1
Resistance of the primary circuit	0.03 Ohm
Capacity of the condenser battery	43.75 micro farads
Resistance of the secondary circuit	7.10^{-5} Ohm
Coefficient of self-inductance of the secondary circuit	4.10^{-7} to 10^{-6}

With $\alpha = 8$, $\omega = 38$, and $\delta = 38.5$, the following expressions obtain :

$$\frac{i_1}{V_0} = -0.046\ e^{-8t} + 3.95 e^{-38.5t} \sin(38t + 40')$$

$$\frac{i_2}{V_0} = 25.8\ e^{-8t} - 33.2\ e^{-38.5t} \sin(38t + 4^\circ 30')$$

In Fig. 1 these values are compared with the corresponding actual data obtained with the use of an oscillograph. The agreement between theoretical and actual values is seen to be of the order of 25 per cent. This must be considered satisfactory, considering that the computation was based on the assumption of a constant permeability of the core and of constant core losses, while the condenser losses were also neglected, the accuracy in charting the curves being some 15 per cent at the most.

TECHNICAL NEWS

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- PERSONAL
- NOTES

NEW EQUIPMENT

V.D. TYPE SWITCHGEAR

A new brochure No. S. 34 illustrates and describes the type V.D. switchgear made by Cooke & Ferguson Ltd., Openshaw, Manchester. This switchgear consists of a short-circuit tested, low-oil content, arc-controlled single break circuit breaker, arranged for vertical isolation in a compact assembly having a small space factor. The rating is up to 350 M.V.A. at 11kV and for normal currents of 400 or 800 A.

Each switch unit is complete on its own transportation truck which incorporates the circuit breaker raising and lowering gear. Lockable shutters are fitted to the housing for shielding all stationary isolating contacts when the breaker is isolated.

The circuit breaker main and arcing contacts (Patent applied for) are of the multi-segment, tulip type with self-aligning high initial pressure line contacts. Positive arc-extinction is effected by compensated cross jet pots. The sliding contacts are spring-loaded and multi-point, and the entire contact assembly is arranged almost horizontally under the breaker top plate. Venting is through a pebble fitted baffle chamber. For earthing, a simple plug-in device can be supplied to effect cable earthing by using the breaker as a non-automatic earthing switch.

To suit requirements, the breaker may be fitted with interchangeable manual, solenoid, spring, or pneumatic operating mechanisms, and a comprehensive range of interlocks is provided to ensure correct sequence and safety in operation, inspection, and maintenance.

Type D. switchgear is available up to a rating of 1000 M.V.A. at 11kV, and Type E has been developed for indoor and outdoor use up to a rating of 1500 M.V.A. at 33 kV in low oil content or in airblast forms.

Copies of the brochure are available on application.

NEW TYPE OF MICROMAX ELECTRIC CONTROL.

A NEW type of M.E.C. has just been announced by Leeds & Northrup Company. Called "Micromax Electric Control, Duration-Adjusting Type," this system not only regulates electric input to hold temperature at a selected control point or to a program, but is said to hold it there so dependably that users obtain the most efficient balance among product uniformity, speed of output, flexibility of operation.

By means of an "on-off" contacting system, it feeds to the heating unit the electric current needed to keep temperature to the required control point or program. Input is either "full-on" or "full-off." Regulation is obtained by controlling automatically the time during which current is "on" and "off."

Employing the basically-sound balance method both for measurement and control, this new system provides full proportional *time* action, as compared with the more familiar full proportional *positioning* action. It provides automatic droop-correction. It can be equipped with Overshoot Control, valuable when the furnace or other unit is coming up to temperature.

To obtain more complete information about this Duration-Adjusting Type of M.E.C., for the temperature regulation of electrically-heated units, write Leeds & Northrup Company, 4934, Stenton Avenue, Philadelphia 44, Pa.

CATALOGUES RECEIVED

Guarding of Drilling Machine Chucks and Spindles. A leaflet issued by Weathershields, Ltd., Birmingham, gives an illustration and description of a safety device manufactured by this firm, called the "Chuk-Gard," for protecting operators of drilling machines against accidental contact with rotating chucks and spindles. Features are compactness of design and easy and simple manipulation so that working speed and efficiency are not interfered with. When closed the appliance completely embraces the chuck and spindle and can be opened in an instant to give convenient access for drill changing or adjustments.

POST-WAR EXPORT.

Czechoslovakian, 34, well educated, at present in Sales organization of large British industrial concern. Perfect English, French, Czech and German, wishes contact with British firm for whom he could utilize his excellent continental connections in building up export of British engineering products. BOX 7007, c/o "*Engineers' Digest.*"

MIDDLE EAST.

Importers of Machinery, Motors, Tools and Technical Appliances, good connections with the engineering industries in Palestine, Syria, Cyprus, Iraq and Iran, wishes Representation of British firms. Technolloyd, 20, Jaffa Road, Tel-Aviv, Palestine.

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CATALOGUES RECEIVED

George Cohen Sons & Co. Ltd., of Wood Lane, London, W.12, and Stanningley, nr. Leeds, have just acquired the whole of the Plant and Machinery of Suncole Works, Cinderhill, Nottingham. This plant was put down new in 1938 and only worked a few months. Designed for the production of solid fuel, smokeless briquettes and bye-products from coal, it comprises Rotary Retorts, Distillation Plant, Gas Producers, very large Conveyors, Aerial Ropeway, two 500 kW. Turbo Sets, two 25,000 lbs. Babcock Boilers, Sidings, and a considerable quantity of Plant of a general nature. Everything is for disposal and a catalogue is available on application.

Eliminates Fractures due to vibration.—A leaflet just published by Clayflex Ltd., Tiddington Road, Stratford-on-Avon, gives an illustrated description of the Clayflex (Patented) Flexible Pipe Union used for eliminating fractures due to vibration. This union is claimed to withstand pressure up to 1500 p.s.i. and above this figure. A synthetic rubber gasket is used in the joint to provide a leak-proof, positive location of the pipe. Gaskets are available from various compounds to withstand all types of liquids, oils, acids, including hot paraffin and steam.

BETTER BRONZE FOR INDUSTRY.

BRONZE with greatly increased strength and toughness can be made from common grades of tin and copper without the purchase of new or special equipment. The process is the result of several years of research and is now disclosed in the two latest publications of the Tin Research Institute, Nos. 120 and 121, which are reprints from the *Journal of the Institute of Metals* (Vol. 70, 1944, 127-147 and 275-289).

Publication No. 120, by Dr. W. T. Pell-Walpole, describes the development of a flux de-gassing process which enables tin bronzes and phosphor bronze to be produced successfully from ordinary commercial grades of metal, or from scrap, where previously high grade metal was essential. This is achieved by using a simple flux containing borax, sand and copper oxide. Bronzes containing 10 per cent tin, suitably cast, can be extruded and subsequently cold rolled and cold drawn. For example, a 10 per cent tin phosphor-bronze gave 24 to 28 tons per sq. in. tensile and 10 to 20% elongation on 2 in. The specification of the comparable commercial alloy (2B8) calls for only 16 tons tensile and 1½ to 4% elongation.

75-Ton Tin-Bronze.

Bronzes with 10 to 14 per cent tin, when made by the new method, can be hot or cold rolled, and the wrought products have exceptional mechanical properties. A 10% tin bronze, extruded and cold rolled, gave 75 tons tensile with 18 per cent extension.

The method of casting giving the maximum tensile and rolling properties is described in Publication No. 121, by Dr. W. T. Pell-Walpole and Dr. V. Kondic. The authors show that variation in casting procedure has a considerable effect on the quality of de-gassed chill-cast 10% tin bronze. By adopting the procedure recommended, a bronze which can be extruded, hot-rolled, forged or hot-stamped is readily obtained. Such bronze will meet many long-felt needs in the engineering world, and bronze bars or strip which can be worked up to 75 tons tensile and yet retain their toughness provide what is virtually a new product with a wide potential field of usefulness.

PERSONAL.

Mr. H. N. Bates, M.I.A.E., M.I.Mech.E., chairman and managing director of Specialloid Ltd., has been appointed to similar positions with the Aero Piston Ring Co. Ltd.

Mr. Norman Clark has been elected director of Darwins Ltd., Andrews Toledo Ltd., and the Wardsend Steel Co. Ltd.

Mr. H. H. Harley, C.B.E., M.I.P.E., has been re-elected President; **Mr. F. W. Halliwell, M.I.Mech.E., M.I.P.E.,** chairman; **Mr. H. S. Holdon,** vice-chairman; **Captain C. R. Cook,** honorary treasurer of The Gauge and Tool Makers' Association.

Mr. E. T. Judge, chief technical engineer of Dorman, Long & Co. Ltd., has been appointed a special director of the Company.

Mr. R. G. Mills has joined the board of the New Insulation Co. Ltd.

Mr. D. A. Oliver, Research Director of William Jessop & Sons Ltd., and J. J. Saville & Co. Ltd., has been appointed also Director of Research to the Birmingham Small Arms group of companies.

Mr. P. Pritchard, M.I.A.E., F.R.Ae.S., has been re-elected President; and **Dr. Harold Hartley,** chairman of the Council of the British Cast Iron Research Association.

Mr. W. P. Ross has been appointed manager of the Dumbarton Works of Babcock & Wilcox Ltd.

Dr. J. F. Shannon, M.I.Mech.E., has joined the gas turbine engineering department of the Metropolitan-Vickers Electrical Co. Ltd.

Sir Llewellyn T. G. Soulsby, managing director of Mountstuart Dry Docks Ltd., has been elected director of Stothert and Pitt Ltd.

OBITUARY.

Mr. Colin Shaw, chief engineer of Firth-Vickers Stainless Steels Ltd., Sheffield, died recently at the age of 49.

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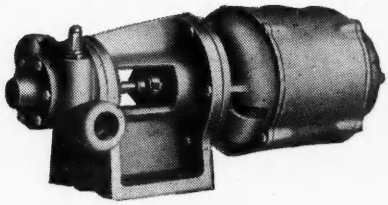

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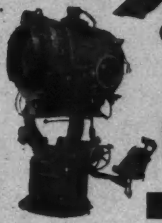

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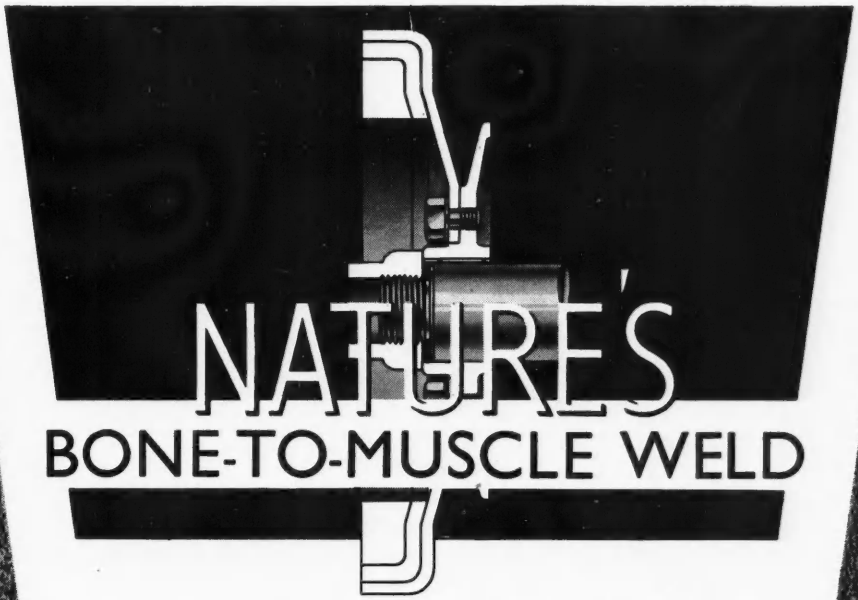

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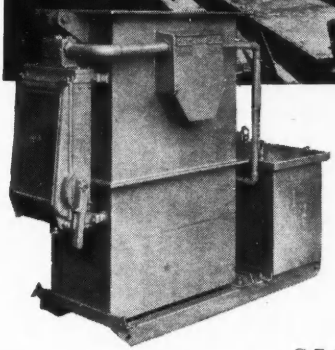

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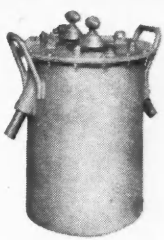

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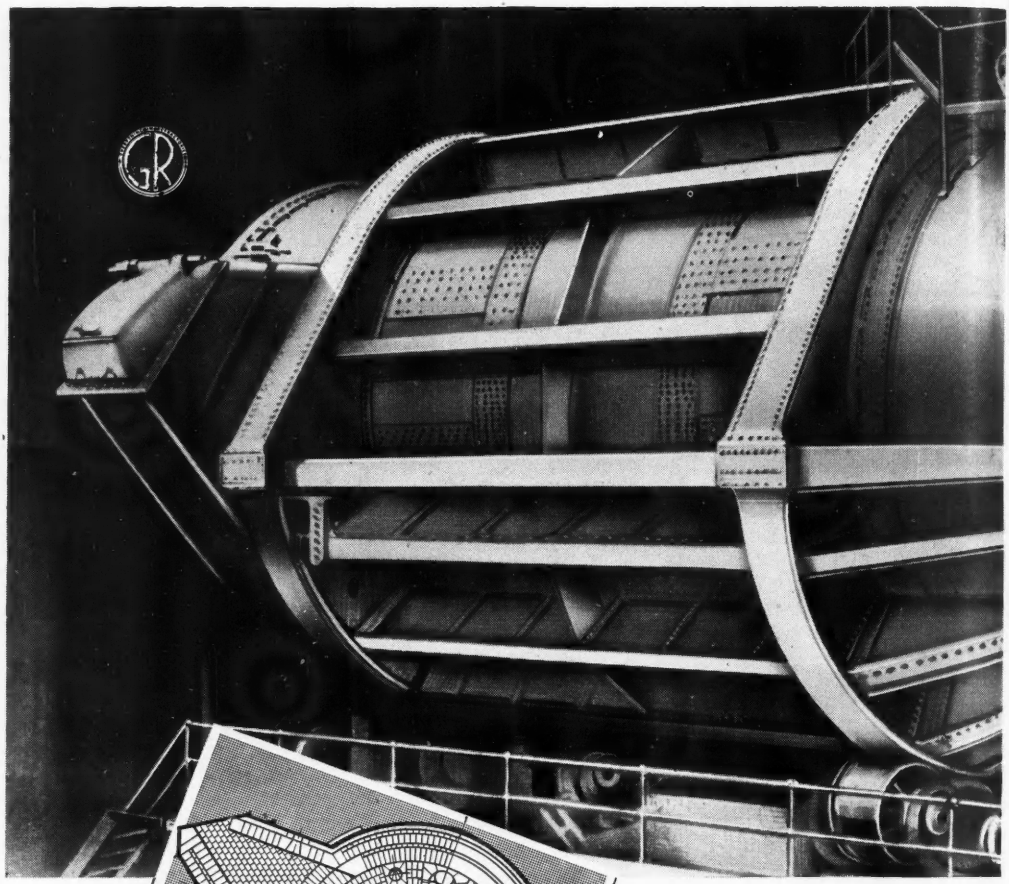

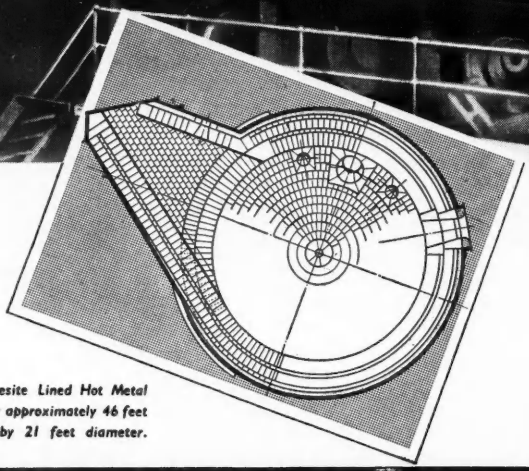

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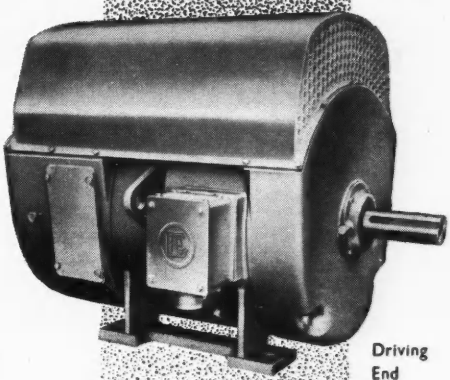

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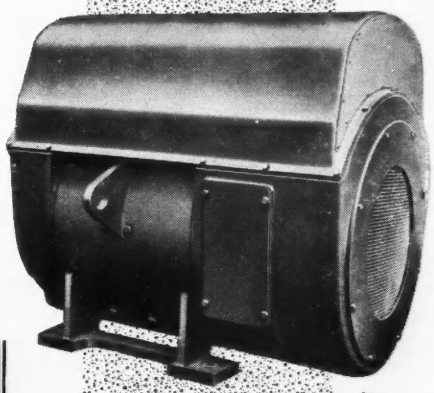

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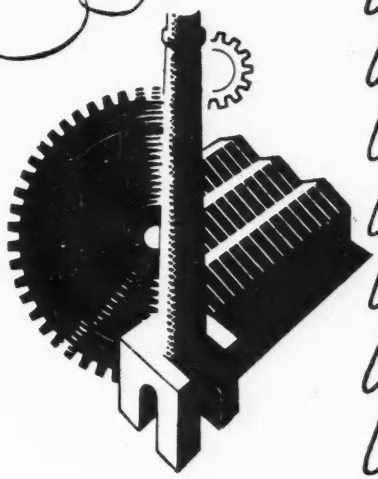

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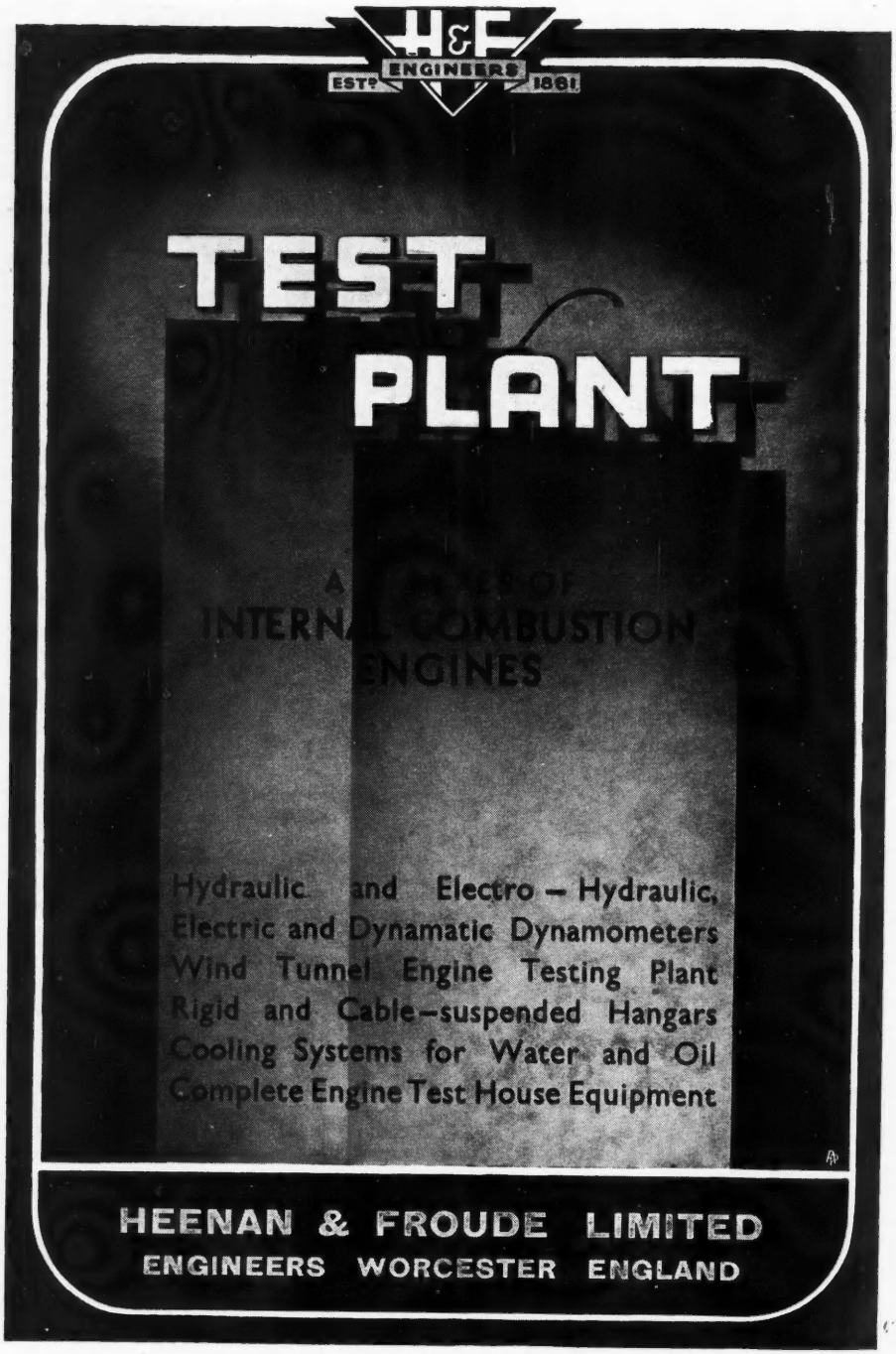
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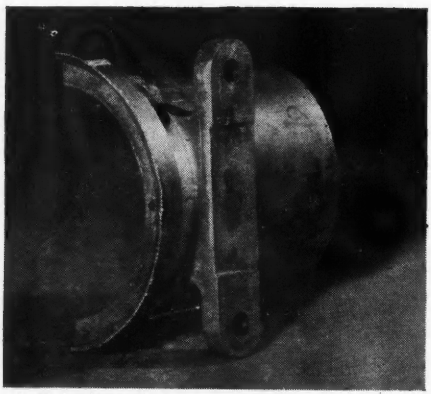
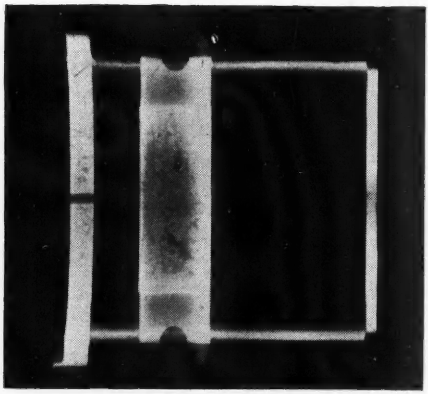
To the engineer "inside knowledge" is as important as to the stock-broker: but, fortunately for the engineer this inside information is easier to obtain and, above all, more reliable. "IndustreX" and "CrystalleX" X-ray films are always at his disposal to provide him at short notice with the 'inside' information about the soundness or otherwise of his castings. From such information faulty castings may be discarded before expensive machining, and any necessary modification to design or foundry technique may be deduced. Often radiography is the only means of obtaining this information.



Photograph of light-alloy casting.



Radiograph of the same casting, showing porosity.

'IndustreX' Type D film—for the routine inspection of light-alloy castings or, with lead screens, for the critical examination of heavy-metal castings.

'IndustreX' Type S film—with fluorescent screens, for heavy-metal castings or welds: without screens, for light-alloy castings with wide thickness ranges, if single exposures only are warranted.

'CrystalleX' film—for the critical examination of light-alloy castings or, with lead screens, for the critical examination of heavy-metal castings.

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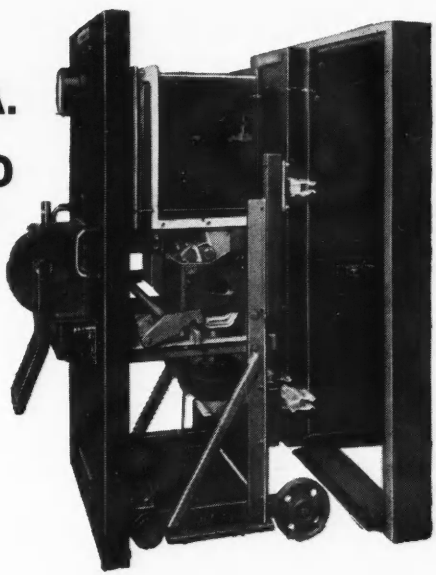

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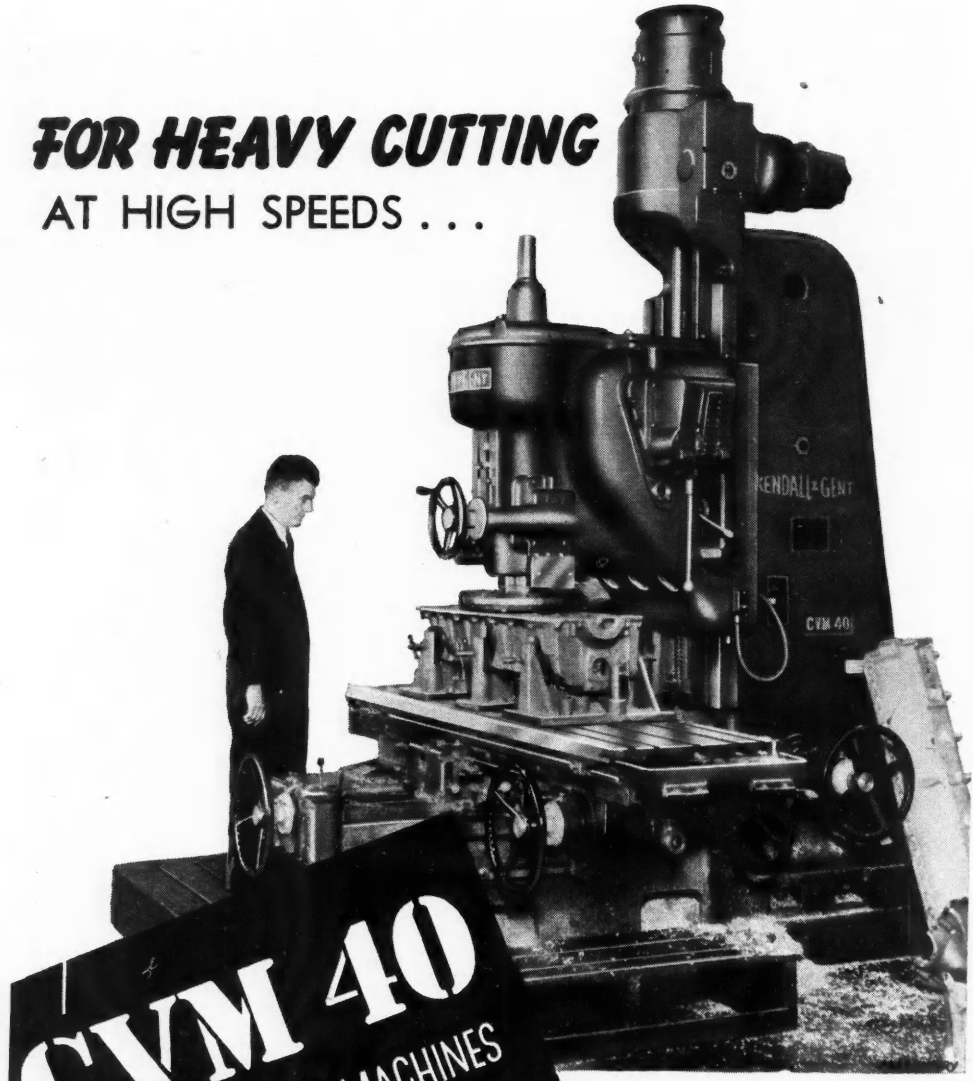

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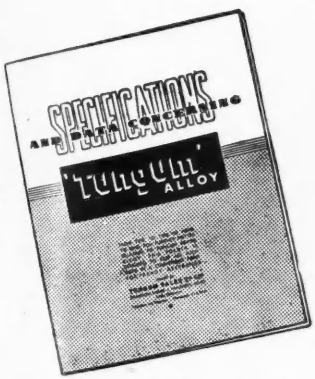

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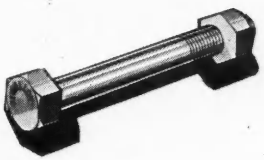
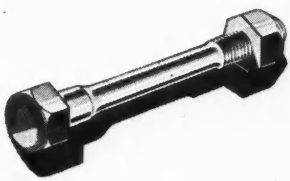
A message to Engineer Designers





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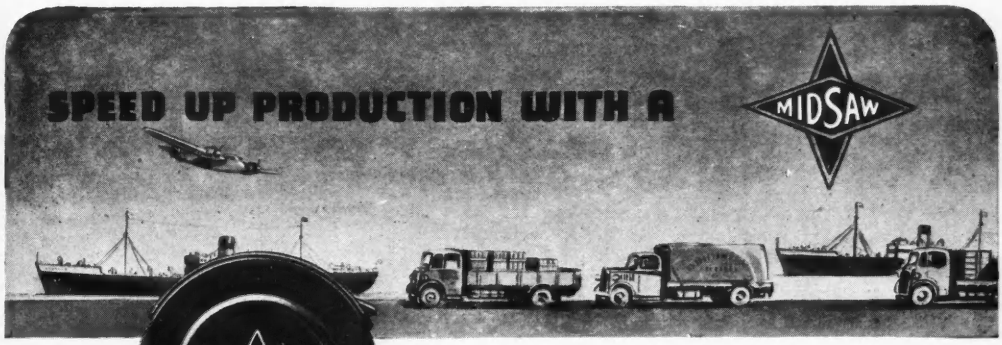

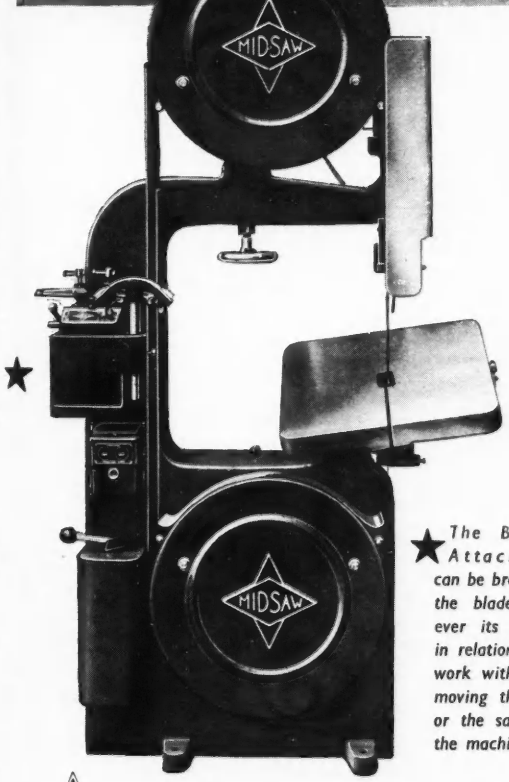

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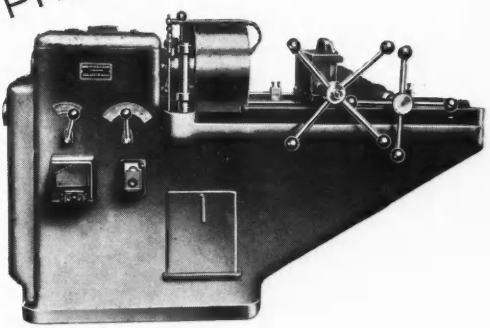

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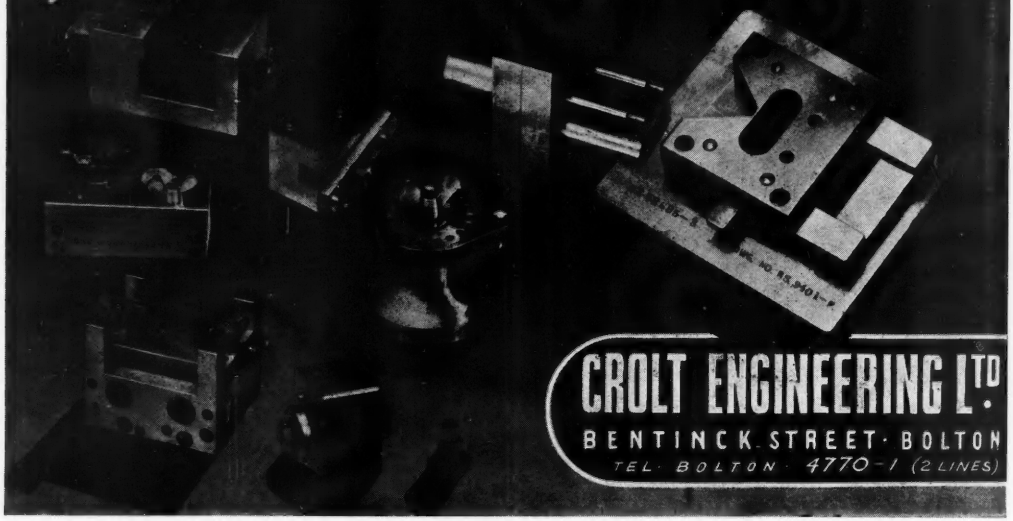

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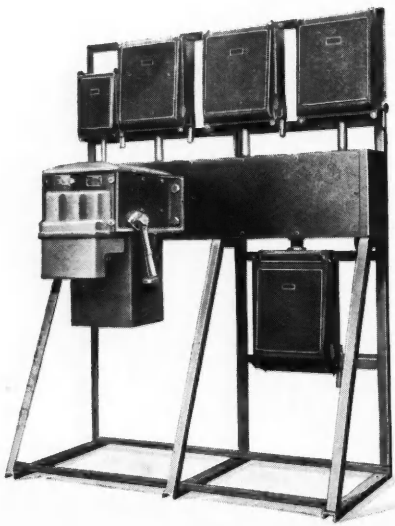

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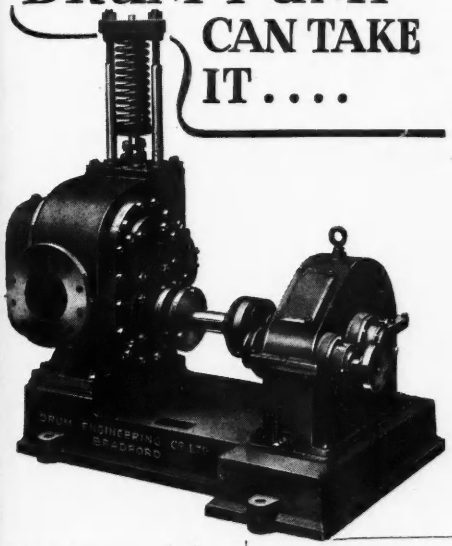

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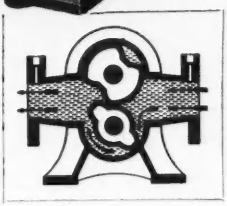

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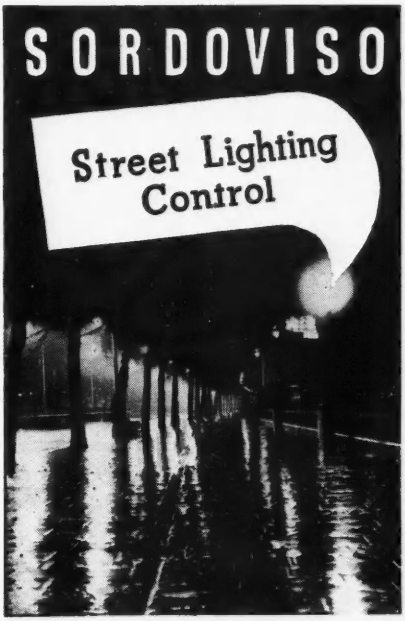

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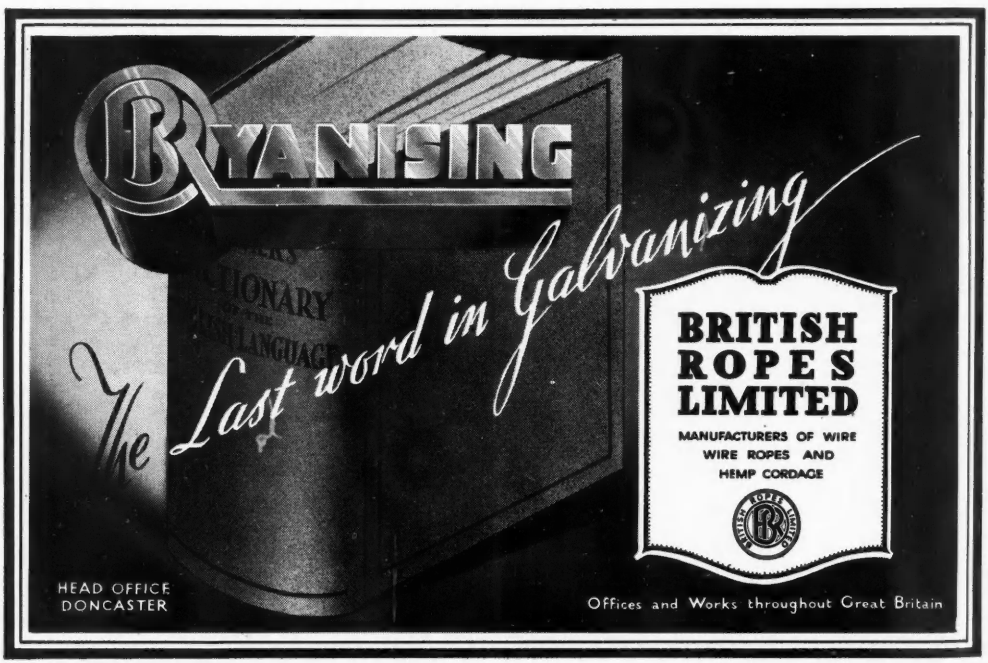
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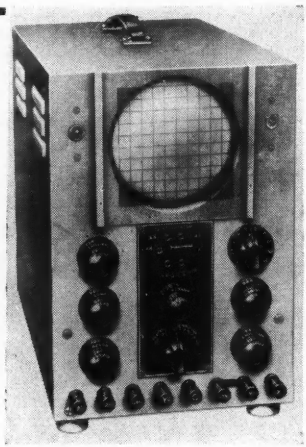

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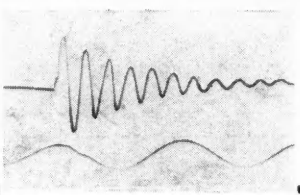

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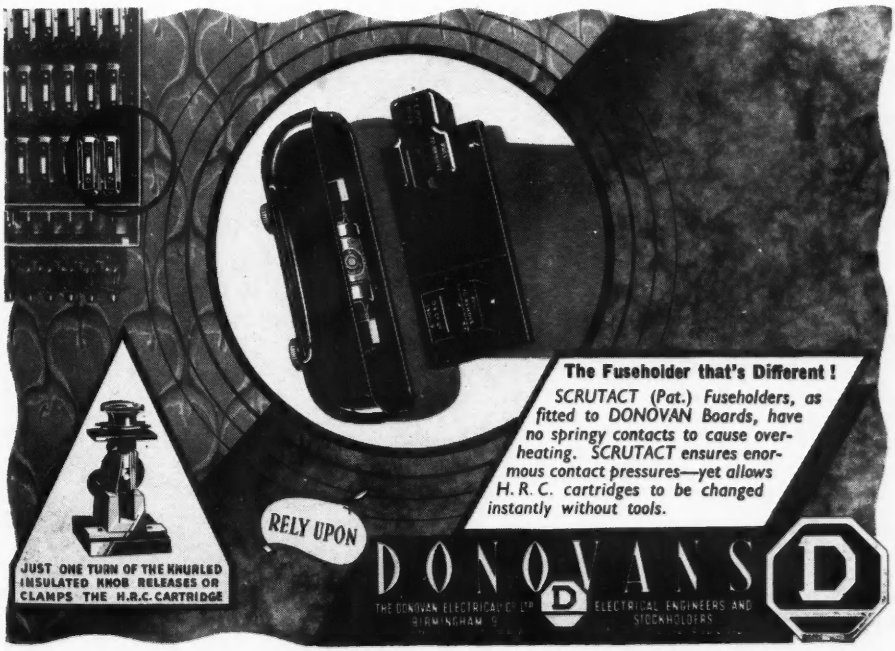
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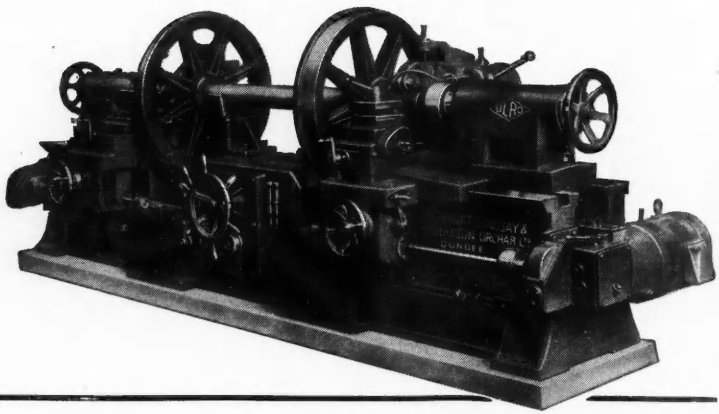

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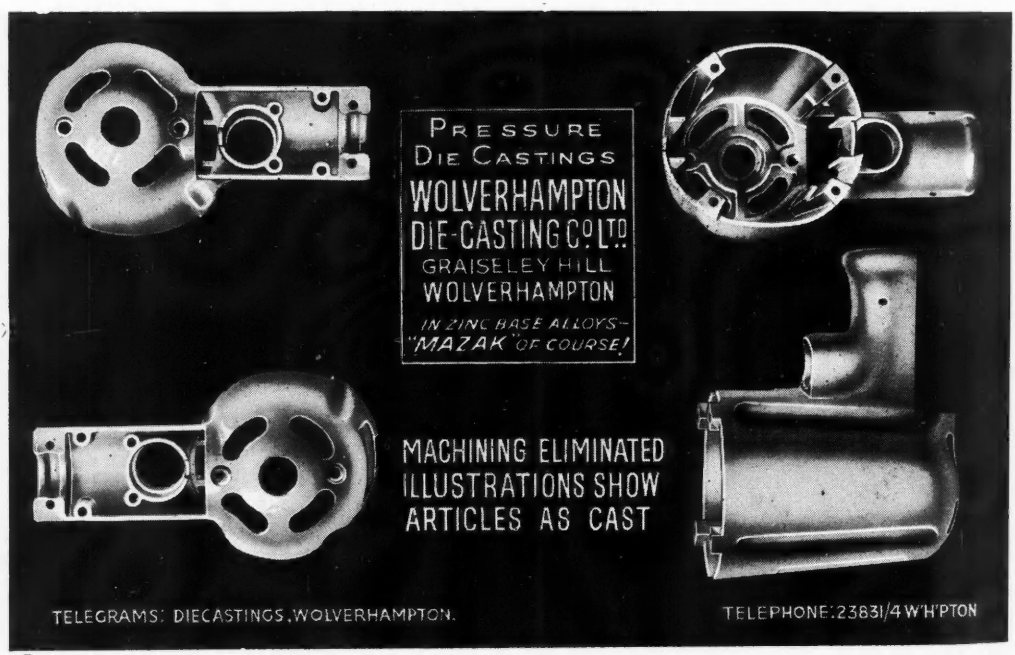

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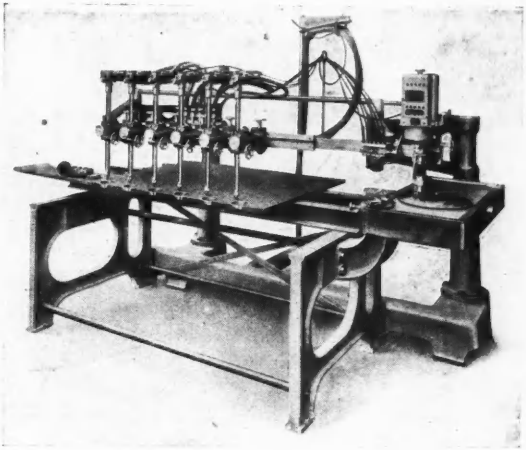

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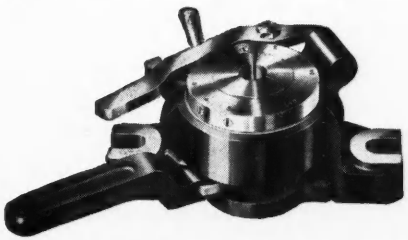

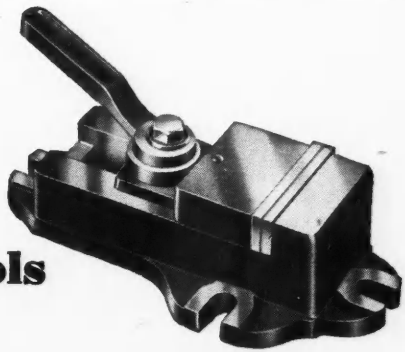

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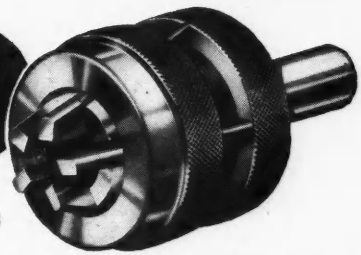

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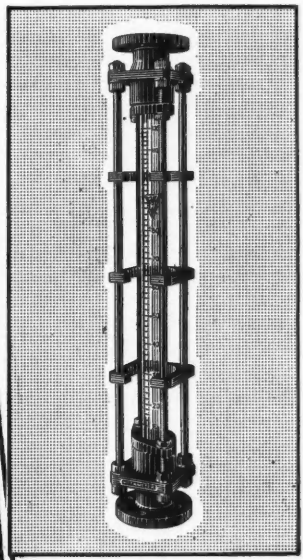

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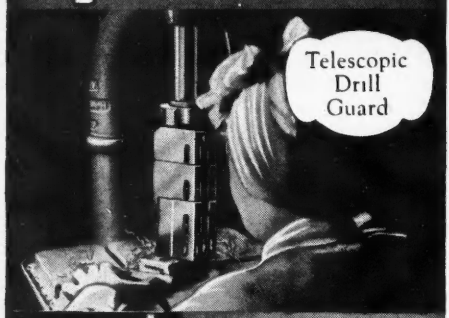

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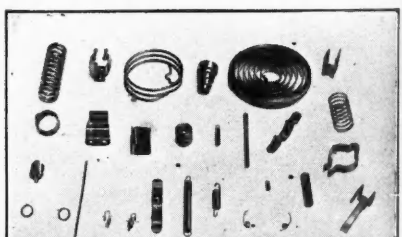

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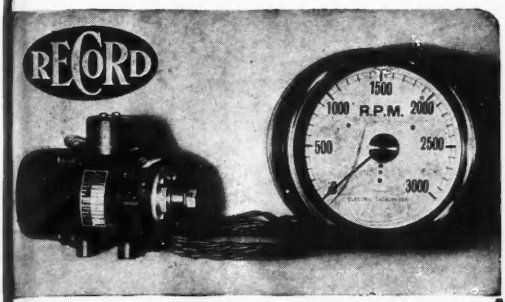

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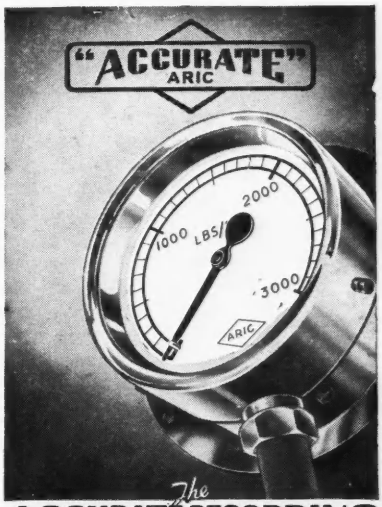

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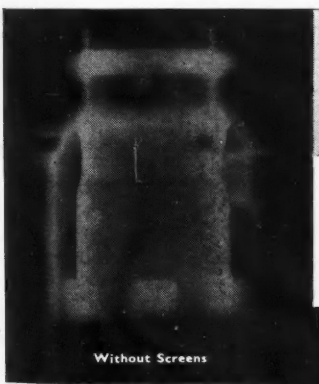

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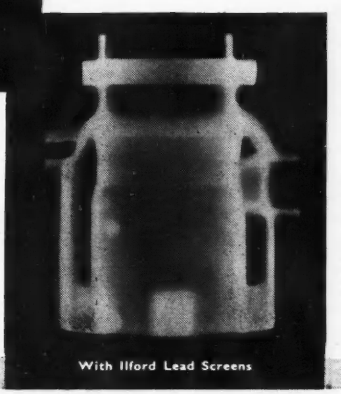

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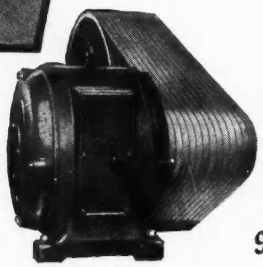

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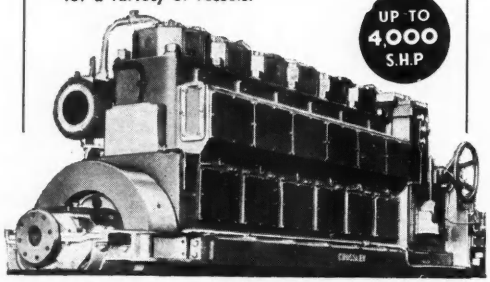

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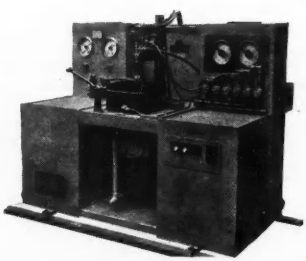

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